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1961 **Rear Admiral E. H. Thiele, U.S.C.G.**

*Names in italics deceased

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JOURNAL

of

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VOLUME 73 AUGUST 1961 NUMBER 3

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VOLUME 73 AUGUST 1961 NUMBER 3

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NAVAL ENGINEERING

Definition: NAVAL ENGINEERING is the art and science applied in the design, construction, operation and maintenance of naval ships and their installed equipment.

SECRETARY'S NOTES

NEW ASNE PRESIDENT

On 5 July 1961, following his retirement from active duty in the U. S. Coast Guard, the ASNE President, Rear Admiral E. H. Thiele, USCG moved from Washington and submitted his resignation. The Council appointed Rear Admiral M. J. Lawrence, USN as President to complete the unexpired term. Admiral Lawrence had just been detached from the Bureau of Ships where he was an Assistant Chief of Bureau to The Secretary of the Navy's Office as Director of Industrial Relations. In his new billet he relieved Rear Admiral R. E. Cronin, USN, a past-President of the Society.

Council Changes

Admiral Lawrence's elevation to the Presidency left a vacancy in the Council. This was filled by appointing Commander Joe W. Thornbury, USN to fill the unexpended term which expires on 31 December 1961. Fortunately, Commander Thornbury will continue to serve as Assistant Secretary-Treasurer.

New Admirals

Action of a Navy Selection Board, selected four Captains, Engineering Duty for promotion to Rear Admiral. All are members of the Society. They are:

Captain E. H. Batcheller who was a member of the Council in 1959-60

Captain W. A. Brockett

Captain E. J. Fahy who is now serving as a member of the Council, and

Captain J. J. Fee

Rear Admiral Robert E. Cronin, USN, Retired

On 1 July 1961, Admiral Cronin who was President of the Society in 1960 retired after 36 years of active naval service. At a ceremony preceding his retirement, Admiral Cronin received the Legion of Merit from Under-Secretary of the Navy Paul B. Fay, Jr., for his improvements in Navy labor relations. Admiral Cronin is joining the technical staff of The George Washington University in Washington, D. C.

Rear Admiral E. H. Thiele, USCG, Retired

On 1 July 1961 Admiral Thiele, President of the Society retired after 30 years of active coast guard service, Admiral Thiele has moved to Cleveland, Ohio as Vice President—Sales of the Cleveland Stevedore Company.

Local Chapters

As of this writing the Society has only the one local chapter which was announced in the May issue of the Journal. This chapter called the FLAGSHIP SECTION was formed in the Washington-Baltimore area with 36 charter members. It is sincerely hoped that the section of the Journal just ahead of Association Notes which will report on the activities of local chapters will expand rapidly and greatly to embrace many more chapters throughout the United States. Potential members are strategically located in all maritime and many inland centers. Each of these require only one determined leader to emulate Mr. Uriah Allen in forming a local chapter. For ASNE activities in the field see page 625.

Errata

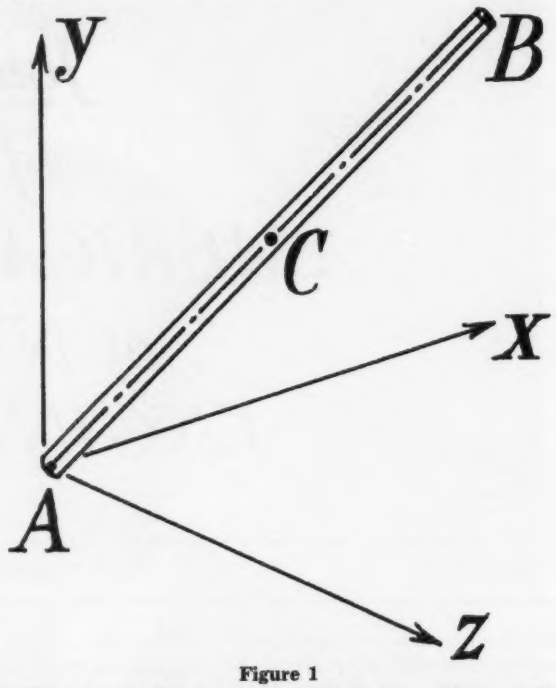
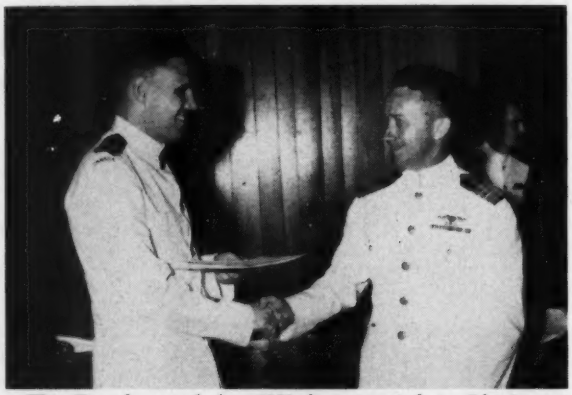


Figure 1

The above figure was omitted from the article "Some Formulas for Piping Design" by Professor John E. Brock, published on pages 395-7 of the Journal for May 1961.

AWARDS

The Brand Award



The Brand award for 1961 is presented to Lieutenant Broughton, R.C.N. by Captain E. S. Arentzen, USN, Professor of Naval Construction at M. I. T.

The Brand Award which was established by Mrs. C. L. Brand in the memory of her husband, Admiral Brand, President of the Society in 1944 was won in 1961 by

Lieutenant W. J. Broughton, R.C.N.

Lieutenant Broughton stood one in Course XIIIA with marks which exceed any previously made by any student. For the first time, the Brand award is carried across the friendly border to the north.

A.S.N.E. Service Academy Awards

U. S. Naval Academy

Official U. S. Navy Photo

The ASNE award for 1961 presented to Midshipman Butrovick by Rear Admiral R. K. James, USN, Chief of the Bureau of Ships and a member of ASNE.

The ASNE Service Academy Award for 1961 was won by

Midshipman Richard M. Butrovick

of Newbury Park, California. Midshipman Butrovick was also awarded the Class of 1924, U.S.N.A. Prize for standing highest in the course in Engineering; the Armed Forces Communications and Electronics Association Award for standing highest in the Electronics portions of the Electrical Science course; and the National Society U.S. Daughters of 1812 Prize for standing highest in the required Electrical Science Courses. Ensign Butrovick is now on duty at a Navy Nuclear Power School.

U. S. Coast Guard Academy

Official U. S. Coast Guard Photo

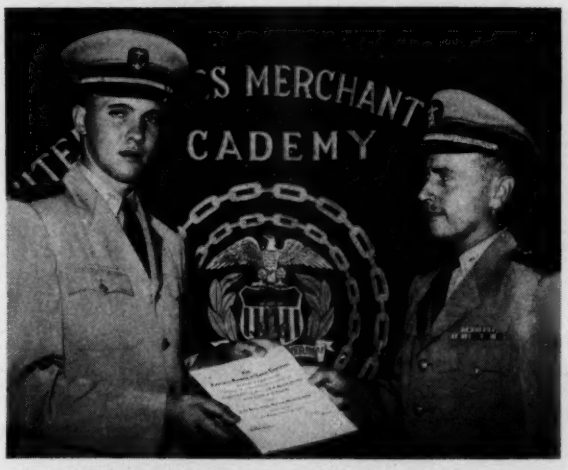
The ASNE award for 1961 presented to Cadet Feldman by Rear Admiral E. H. Thiele, U. S. C. G., Engineer-in-Chief of the Coast Guard and President of the Society.

The ASNE Service Academy Award for 1961 was won by

Cadet Donald A. Feldman

of Seattle, Washington. In 1957, Ensign Feldman accepted appointment to the Coast Guard Academy in preference to the Naval Academy, Military Academy, Naval ROTC and the Dow Chemical Engineering scholarship, all of which were offered to him. He led his class academically for four years and was a leader in extra-curricular activities. Upon graduation he was honorman in his class of 119 members with a four-year academic average of 94.24. Of 33 graduation awards presented to his class, Ensign Feldman received 16. He is now stationed on the Coast Guard Cutter *Klamath*.

U. S. Merchant Marine Academy



The ASNE award for 1961 presented to Cadet Dudes by Captain D. Furlong, U.S.N., member of ASNE, Planning Officer at New York Naval Shipyard.

The ASNE Service Academy award for 1961 was won by

Cadet Anthony L. Dudes, Jr.

of Newark, New Jersey. Cadet Dudes ranked third in his class of 111 engineers with a course average of 3.41. He was selected for the ASNE award because of his standing in selected engineering subjects and of his potential as a Marine Engineer.

Advertising in the Journal in 1961 has fallen far below budget estimates. Legislative action which is pending may reduce the Society's advertising income still farther.

TO AVERT AN INCREASE IN DUES
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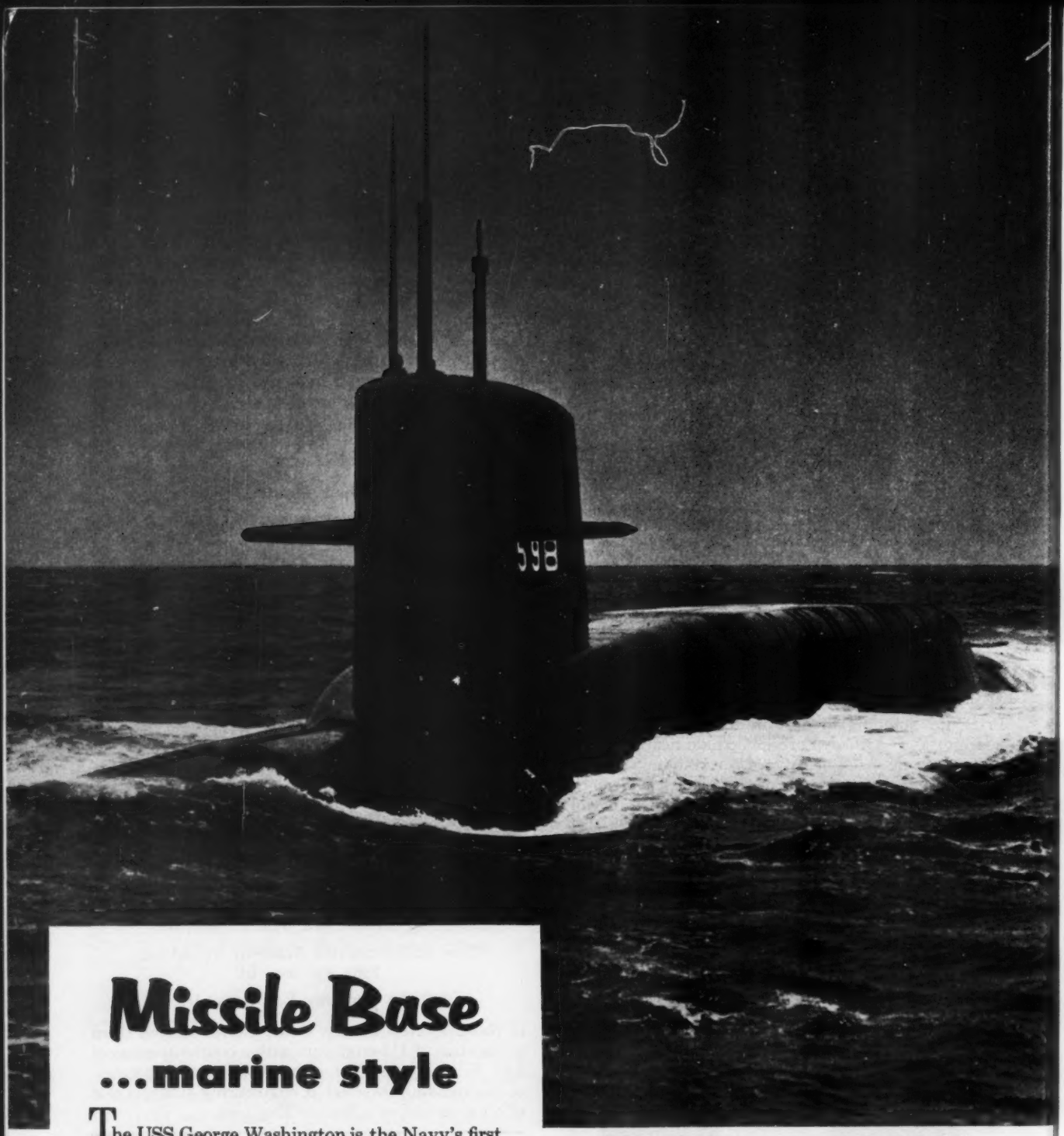
will recruit a
NEW MEMBER



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OFFICIAL U.S. NAVY PHOTO

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CAPTAIN JOHN ADRIAN HACK, UNITED STATES NAVY.

NAVAL MANAGEMENT IN THE AMERICAN CULTURE

THE AUTHOR

is a graduate of the Naval Academy in the class of 1935, and a graduate of Rensselaer Polytechnical Institute in the field of Management. He has had a varied career in the Fleet, having served in battleships, destroyers, submarines and the amphibious forces. He has had staff assignments with Commander Submarine Force Asiatic Fleet, Commander Submarine Force, Commander Cruiser Destroyer Force, in assignments which covered personnel, training, logistics, operations, and communications. A graduate of the Naval War College, and the former Director of the U.S. Navy Management School, he is at the present time Commander Amphibious Squadron Seven. This is the Eighth article in which the fictional character of Captain O'Shaughnessy has appeared, the last being "Captain O'Shaughnessy's Oath" in the August 1960 Journal.

WHAT THE SUCCESSFUL EXECUTIVE DOES

EVEN THOSE WHO never read the sport pages recognize the name, Casey Stengel. Yankee lovers, Yankee haters, the man in the street . . . even Elizabeth Bridget O'Shaughnessy . . . know him as one of baseball's most successful managers. Why? Because in a golden decade of New York baseball, he dominated the American League! He lived by the same basic principles that every man must follow, if he is to be a successful naval executive.

When Mike and Liz were first married life was serene. He was an ardent Red Sox fan, having been raised in the shadows of Fenway Park. Liz had no more intimate knowledge of baseball than she would have acquired, had she been a Buddhist monk. One spring she had gone to Florida with another Navy wife. She came back, not only an avid baseball fan, but an expert as well. Elizabeth had actually *met* Ole Case!

"Mike, you should have heard him! No wonder the Yankees always win with a manager like that. He's a leader! Why that first day he got the players together and said:

"BOYS! We have got to win 100 games in those eight American League parks this spring and summer so we can get into the World Series again. I've sort of gotten used to that extra dough in October. Don't really need it, with that bank I've got out in Glendale, but I'd hate to think that Yogi, Mickey, Whitey and the rest of you wouldn't get it this year. Now we've got some real good coaches who'll help you. But you're Yankees! You do the ball playing. And leave the masterminding to me."

As each season progressed, Liz O'Shaughnessy followed the fortunes of the Yankees as though she were a part owner. When they went into a slump she'd worry. She could almost see him looking over the performance charts of the players, examining the schedule, analyzing what went wrong that he hadn't anticipated. She watched from afar, through the sport pages. Liz could remember why and when he brought up Rhyne Duren from Denver; the year he bought Hector Lopez from Kansas City. She watched him juggle the lineup and play the percentages. And in October blue eyes would sparkle.

"Oh Mike, isn't he wonderful, he did it again! Do you know why? From that first day in Florida he set his sights on the pennant! Casey knew what had to be done too, and so did all the Yankees. When they lost all those games in July he didn't panic. No sir! Remember, he benched Mickey Mantle, moved Berra to right field, and put Howard behind the plate. He wasn't making emotional decisions. Casey's as coldly analytical as a loan agent. That team plays as though every man knew exactly what the other fellow was thinking, and what he would do. Did you ever see three Yankees colliding over a fly ball? No wonder they won. Casey Stengle isn't afraid to manage that team. He makes decisions, the team knows baseball better than the umpires, and every Yankee thinks of himself as a star. He has the illusion that nobody, even Stengel himself, is as good as he is!"

Captain O'Shaughnessy hadn't thought too much about it until his red head had captured him in her enthusiasm. Liz had another effect on him. Until he had been subjected to Elizabeth's outbursts over baseball, something which should have been very obvious to a naval officer, wasn't obvious at all. The word "manager" had a connotation of logistics to him. Yet Liz had made it very clear even to him that Casey Stengel was a successful baseball *manager*, for much the same reason that he might be a successful naval officer. . . . . Casey was a leader. He had enthusiasm for his profession. He had acquired a wealth of experience through the years. But equally important, and perhaps what set him apart, was that he directed the Yankee baseball team by a proven set of principles. He set an objective; devised a plan which if carried out would allow his team to win the pennant; analyzed events as they occurred and took prompt action to correct faulty performance which endangered his objective; gave positive and clearcut direction; taught fundamentals so thoroughly that each Yankee played the game in the same manner, no matter how the lineup was juggled; and somehow he made each player think of himself as a star. What Casey did, a Naval Officer must do. So must every leader, whether he be an executive in General Motors, the Chief of Naval Operations, the Commander of a ship or shore station, a baseball manager . . . . . or *you!* The only difference will lie in the sound of the words which describe the objective; the scheduled actions which comprise the plan; the complexity of measuring actual performance and comparing it to the standard; and the magnitude of the decisions which must be made.

This seemed obvious and elementary to Captain Michael O'Shaughnessy, when he stopped to think about it. The art of management *is* very simple for the man who has the courage and the conviction to adopt certain proven principles as a way of life, and as a philosophy of doing. But each of us, unwittingly become so enthralled with prejudices and biased beliefs which experience *seems* to have taught us, that we rely on guesses and expedients to the bitter end . . . until the ship goes down. When that happens, it doesn't make us feel any better to know that some well known men, and some famous baseball names, have foundered on the same reefs which surround those who will not be guided by principle.

Whatever your present position, challenge your performance against a review of some of these basic signposts to success. Don't follow them blindly. Do let them stir questions and discussions in your mind.

THE OBJECTIVE

What *is* your objective? What do you want your organization to achieve this year? This month? This week? Tomorrow? If you don't know how will you know when you get there? If you haven't formally told those upon whom you rely to assist you, how can they act intelligently to insure that the common goal is reached?

Except at the very top echelon (and if you are at that point you have no need to further develop your executive talents, for you are already a past master), the objective is received from an immediate superior, although sometimes it must be assumed. Within the military it is many times expressed as a mission, that is, a task to be performed and the reason therefor. Many times it is received as a directive, or as a policy. A time element may be expressed in order to insure overall coordination. Obviously objectives will be very broad at the Joint Chief level, and will become more specific as one descends the organizational ladder. There is a spectrum through which objectives run . . . . . from a broad strategy, through a terse directive as to what is to be done, to a more detailed exposition at the lowest level. To the Admiral the objective sounds like "what". To the seaman apprentice it sounds more like "how". Whatever the level objectives must be meaningful. Whether they be broad or specific, they must be so clear that there can be no misunderstanding them. *Heaven* is a thrilling goal for which to strive. Beware of the preacher who implores you to reach for *Eternity*. There will be surprises awaiting the careless and misguided.

THE PLAN

What actions must be taken in order to achieve

the objective? *Who* must perform them? *When* must these be accomplished? *Why* are they necessary? *Where* is it necessary that they be done? These are the all essential ingredients of a plan. Collectively they describe *how* the commander intends to have his entire organization work together toward the common goal. Materials must be procured and fabricated into usable ships, weapons and facilities. Personnel must be recruited, distributed and trained. Capital must be raised and expended. Fleets must fight battles. Individuals must shoot guns. All must happen on a coordinated time schedule, which will answer for the Admiral and the seaman, "Who, What, When, Where, Why!". If any of the elements are lacking, the chances for coordinated action by all personnel to strive for the same objective at the proper time will be no better than *chance*. No better than the results which might have been achieved by the flip of a coin! Such a schedule may be recognized by many names . . . . . a budget, a production plan, an operation order, a forecast. Whatever the name a rose is always a rose. It must have the same characteristics.

Planning is one of the most important arts which the naval officer must master. Many times it is one of the most neglected of his efforts. Unwittingly he sometimes fails to gain the staff assistance he needs, because he doesn't fully understand the value of management techniques. In failing to place the word "manager" in its proper perspective, he may sometimes not recognize what techniques are available to him through specialists in various areas of management. The Personnel specialist can assist in establishing a sound organization; the production specialist can assist in setting the time sequence and defining the essential elements of action; inventory specialists can advise in the procurement and distribution of essential materials; operations analysts can advise in the most profitable courses of action; electronic data processers can supply a wealth of building blocks from which a usable timetable of standards can be devised from which to measure deviations; money being the common denominator which describes the relative value of materials, facilities, and manpower, the financial comptroller can offer valued advice; lastly, the human relations specialist can be a most valued teacher. The naval officer executive need not be a specialist in any of these fields. He must have a knowledge of each, however, so well grounded that he will blend their advice into sound plans. One thing he must have. He must possess an enthusiasm, and a confidence which will inspire others to follow him. No plan of itself has ever accomplished a single action. The successful naval executive, no less than Casey Stengel, must motivate his followers to strive for the objective willingly and with zest. Every man must believe himself to be a star of the team!

With an integrated plan of action, well understood by all personnel, how *much* is made routine. There is no question of coordinated effort. Every man knows exactly what he must do, and when he must do it. Every day becomes a positive step toward an assured result. Priority of effort is so clear. Actions assume an importance of the moment rather than that of a crisis or an isolated event. Don't we lose flexibility? How flexible can one be today, if the action required is a key element toward reaching a prescribed goal next year? Flexibility is an expression of time. It is impossible to turn back the clock with all the wishful thinking in the world!

A plan serves as a series of buoys planted along a channel, stretching into the harbor one is seeking. With it an individual leader can check off the events which should happen, with the full assurance that he and his assistants are steering the correct course.

EVALUATION AND CONTROL

Without a plan, all events observed may seem unique and their relation to the ultimate objective obscure. Some may serve to create problems, coming as they do by surprise, like a thief in the night. But viewed against a standard of what should have happened, actions will appear to be buoys passed on schedule, or warnings that the ship is off course, and that the unexpected has happened. In the latter case immediate action is indicated.

The concept of Management by Exception stems from this basic fundamental principle. A naval officer need not worry about those things for which he planned and wanted to happen. It will serve him no useful purpose to collect a torrent of reports, and waste his and other's time in analyzing actions of this type. He may get ulcers. He should get them by worrying about those actions which *did not* happen in the manner predicted, and which require prompt decisions if he is not to end up on a reef. There is one danger in this concept. Top executives have a tendency to select the *critical* events of an operation, and make provision for reporting deviations from these only, failing to realize that deviation from the plan in a *preconceived* non critical area might be serious. It has happened that for want of a nail a battle was lost. *Anything* which is not routine, should be reported to the next echelon. Any non scheduled event should be viewed as being critical to the person who scheduled it, and so reported. It is interesting to conjecture, however, about the usefulness of the torrents of data which flow upstream, describing events which happened exactly as they had been planned.

Human relations become paramount when actions are at variance with the plan. If the objective is still desirable and the plan is still feasible, all the screaming and shouting that "Someone made a mistake" will only serve to delay the decision required to get the ship back on course again. The logical solution will be reached sooner, if all concerned analyze what happened, why the action was at variance with the standard, and what has to be done to

correct the error. Management analysts may help at such a time. Accounting tools are useful. Electronic Data Processing will become more and more one of the most valuable tools yet devised. Electronic machines can rapidly compare what was supposed to happen with what did happen, and give quick, assured clues as to what the problems are and how seriously the objective is endangered. They can do this only if there is an assured standard method of storing the information. Each echelon must be viewing the same information. Each must be storing the information in a compatible manner. A serious danger lies in the fact that some naval executives tend to rely on these electronic marvels to provide answers, instead of problems!

THE DECISION

More has been written about the decision making process than any other area of executive action. Yet most decisions actually require more courage than sagacity. If planning has been done properly, and the available tools employed which allow a comparison of what has happened with what should have happened, there are few choices possible. One decision may be simply to rectify the error and resume the schedule. While this may be easier said than done, careful analysis and prompt action will many times show the way.

It may happen that the only decision we can make is to delay, or to recommend a change in the objective. This is a situation we, as naval officers hope will never happen. The efforts of others at parallel echelons will be affected. Most certainly our seniors will have to take action. We may be tempted "to wish the problem would go away" by ignoring it for the time being. We may, through emotional outbursts and lack of understanding encourage just such an attitude among our juniors. Granted that incompetence and stupidity have no place in the makeup of a naval officer these must be removed. But the fear of making an honest mistake will inevitably stifle initiative. Even worse it will tempt those who are able to avoid the possibility of such an occurrence by packing waste into their planning, thus insuring that money, manpower, materials, facilities and time will never be in short supply.

The type of decisions we make can be only as valid as our objective, as wise as our planning, as honest as our evaluation, and as courageous as our character and convictions. Decisions of our subordinates depend to a large measure upon the environment which we create.

MANAGEMENT IN THE AMERICAN CULTURE

In any culture or society these principles will apply. The executive in Russia, the businessman in Japan, the military officer in Germany all had to follow these guidelines or be doomed to failure. Students of business and history can point to the block over which one famous personality or another stumbled. Of equal importance to us, as naval officers in the United States Navy is *how* one manages. Man must live and work in the culture and environment of his own society. If he attempts to establish an organizational philosophy which opposes the culture in which it operates, he must either overcome the culture, or be doomed to failure.

The American way of life is based upon the premise that individuals are of prime importance in society. We believe that their rights are given to them by a Power which is supreme over the State. In order to protect these inalienable rights individuals, by their consent, form governments. People are not masses, nor are they classes. Legislative tyranny is as alien to us as is dictatorship. Individuals have God-given wants, desires, emotions and talents. They have a God-given responsibility, which they show by the drive of initiative, to use these gifts in a quest for happiness. When executives forget these basic facts, or subtly ignore them, they endanger the success of their organization. Whether the executive manager is in the automobile business, manufacturing electronic goods, or entrusted with the defense of his country, he must remember the truths which our founding fathers described as being self evident. Literally, no proof is required, these ideas are so inborne into man.

THE INGREDIENTS OF MANAGEMENT

One eminent authority of military management described the ingredients of all good executive direction as "The Management Martini." Most people who have even a passing interest in a polite before dinner cocktail would recognize that a Martini consists of gin, vermouth and ice. Some variations of the proportions are acceptable. Two to three parts of gin, with one part of vermouth, stirred in a mixture over two cubes of ice will produce a palatable drink, providing that one likes Martinis. Other variations may well send your guests home reeling!

It is of more than passing interest to note that such authorities as the British Admiralty, the General Motors Corporation and the United States Department of Defense are in complete agreement concerning the ingredients of the Management Martini. Each advocates, for the proper administration of its organization, three basic items . . . . Common Doctrine, Centralized Direction and Decentralized Execution. Mixed in the proper proportions these will insure good management. The trick is to discover the proper proportions for the size, character, diversity and complexity of the operation, and the cultural environment of the individuals who collectively make up that organization. The manuals do not disclose the exact proportions. However we do know that the larger the organization, the more delicate the balance of the ingredients becomes, and

the more temptation exists to mix a bad martini. This as naval officers must concern us, lest we send our guests home reeling!

COMMON DOCTRINE

Common Doctrine, by definition, is a set of common beliefs, tenets, dogmas or principles subscribed to by any organization of people working together in a common cause. With such doctrine, one can reasonably predict and be assured how the group, and each individual, will react to a given policy or directive. Without such common beliefs, no amount of persuasion or wishful thinking will allow coordinated action without a stream of detailed directives and direction. With a man on first and none out, a sign from Casey Stengel to bunt means just one thing to every Yankee player. Casey wants that man to reach second. A simple phrase was sufficient to set the entire submarine force into action at the beginning of World War II. "Conduct unrestricted warfare against the Japanese" was completely understood by every submarine commander in the Pacific. Commander Submarine Force knew exactly how each would react. The simple signal "Land the Landing Force" sets a complex host of Marines and Sailors into a coordinated pattern of actions.

Customs, traditions, training, long standing policies and attitudes combine to form the common doctrines which collectively characterize the organization. The longer they exist the more firmly embedded they become. The larger the organization, the more vital they are to its well being and the coordinated effort of its people. Excellent communications, consistent policy and conformity to the cultural environment are essential to maintain them. Individuals will inevitably resist directives which openly, or insidiously, tend to destroy the things which men hold sacred. In America, for example, we resist Communism principally because it teaches as a doctrine that individuals have no rights, but rather derive their privileges from society as a whole. Any organization will meet the same resistance if a doctrine is taught, or implied, that the individual is less important than the system. The Moral Leadership Program throughout the Navy is based upon individual importance. Naval Officers, probably more than any group outside the military, devote their careers to proving in action that these beliefs are sacred to them. But each of us is tempted many times to forget them. We cannot, and be successful.

CENTRALIZED DIRECTION

Clearcut guidance from each organizational level to the next responsible echelon is essential, if all are to work toward a common goal. The leader must lead!

Direction may take the form of plans, formal objectives, manuals or even oral speeches. The motto of a shipyard "Our only mission is to serve the Fleet" is a form of centralized direction. So is the age old "Now hear this! A clean sweepdown fore and aft!" The important factor is that such direction must be consistent, timely, and conform to the common doctrine which has been established.

There is always the danger that centralized direction may become confused with centralized directives. It is a phenomena that bigness begets directives rather than direction. Admiral King, one of the greatest military leaders of our time, frequently admonished his staff and subordinate commanders "Tell them what we want, not how to do it!" As frequently he warned against the practice of disregarding standard procedures, practices or instructions. Standardization obviously is required "in order that what is to be done shall be done .... as the correlated part of a connected whole."* In King's words:

> "I have been concerned over the increasing tendency now grown to standard practice, of flag officers and other group commanders to issue orders and instructions in which their subordinates are told 'how' as well as 'what' to do to such an extent and in such detail that the 'custom of the service' has virtually become the antithesis of that essential of command—'initiative of the subordinate.'
>
> "If subordinates are deprived of that training and experience which will enable them to act on their own, if they do not know, by constant practice, how to exercise 'initiative of the subordinate'—if they are reluctant to act because they are accustomed to detailed orders and instructions— if they are not habituated to think, to judge and to decide and to act for themselves in their several echelons of command—we shall be in sorry case when the time of 'active operations' arrives.
>
> "The reasons for the current state of affairs are many but there are four which need mention; first, the 'anxiety' of seniors that everything in their commands shall be conducted so correctly and go so smoothly, that none may comment unfavorably; second, those energetic activities of staffs which lead to infringement of the functions for which the lower echelons exist; third, the consequent 'anxiety' of subordinates lest their exercise of initiative, even in their legitimate spheres, should result in their doing something which may prejudice their selection for promotion; fourth, the habit on the one hand and the expectation on the other of 'nursing' and 'being nursed' which lead respectively to that violation of command principles known as orders to obey orders, and to that admission of incapacity or confusion evidenced by 'request instructions.' "**

Naval Executives face one of their greatest dilemmas in this area. Some standardization is required, in order that control may be exercised, and in order that individuals will react in a team effort. As a guideline one might well ask himself, before he signed a directive, "is it really necessary that everyone in the Navy do this action in this exact manner. Is this really an area of Common Doctrine?". If the answer is "no," don't sign it! For the larger the organization, the greater the confusion and unbalance can result from an overdose of such direction. The problem faced by the subordinate may not be the same one which is being faced in the superior's

* CINCLANTFLT Serial 0328 of April 22, 1941.
** CINCLANTFLT Serial 053 of Jan. 21, 1941.

mind. Just as dangerous, such detailed directives conflict with the most firmly embedded of American Common Doctrine, the importance of individual initiative. The ultimate result may be a disregarding of more important direction.

Centralized Direction is the most sensitive of the ingredients. It is the "gin" of the Management Martini.

DECENTRALIZED EXECUTION

The man closest to the scene of action knows best how to solve the problem at hand. The principle of decentralized execution is fundamental to the concept that authority and responsibility are as closely united as man and wife. Our fundamental culture is based upon it, for it emphasises the worth, the dignity and the initiative of the individual. It is one of the corner stones of the organizational philosophy and the strength of the United States Navy.

The commanding officer of a ship at sea has the complete responsibility for the safety and well being of his ship and crew. So great is his authority that he will be courtmartialled for slavishly following rules and regulations which would drive his ship upon the reefs. Faced with the decision of repairing a ship needed by the Fleet or violating a Bureau of Ships directive, a Commander of a Shipyard will probably be scolded if he neglects the ship. Each of us knows how impatient we get with an officer who stubbornly or slavishly sticks to the "letter of the law," knowing that in doing so he is creating disaster.

Centralized Direction is the most sensitive of the Management Martini. Common Doctrine is the ingredient which gives cohesiveness. But Decentralized Execution is the most important ingredient. Battles are won by the men in the ships and planes at sea. Ships are repaired by the workmen on the docks. Only individuals can create the actions which will accomplish the objective.

In the words of Admiral Arleigh Burke:

> "The basic element of military capability is man, individual man, with his personal dignity, and his pride. No matter what machines, what weapons evolve, they will be a product of man. Man will maintain them, and above all, man must control them."*

SUMMARY

The naval officer carries out his responsibilities in a culture which emphasises the dignity, the importance, and the unquenchable initiative of the individual. He is the leader. He must set the goal. His brain formulates the plan which if carried out will assure success. His astuteness allows him to evaluate events and prompts him to make timely and courageous decisions, or if you will to give his direction to the efforts of his subordinates. If his juniors are to be enthusiastic and skillful, he must generate common beliefs among them which will spur them to effective individual action. He is a leader, therefore he is a Manager—a moral manager!

Admittedly the foregoing concepts are simple, yet they must cause each of us to think, to review our own actions, and perhaps improve our value to the Navy in the process. To some extent they are abstract ideas, difficult to translate into our present positions. They are none the less applicable to each of us . . . . . in a Bureau section or division, on a ship, or in one of the many shore establishments.

Perhaps a review of the problems of one family, the experiences of one Bureau, may lend understanding to the ideas. From these we may evolve further fundamental principles which will guide us in our executive actions. As a by product it may help to clarify some erroneous impressions we have acquired concerning planning, budget, and accounting and stimulate an interest in those areas in our day to day management of our part of the Navy.

THE O'SHAUGHNESSY BUDGET

Like so many words used in management studies, the word "budget" brings to mind varied pictures. The American public associates it, as an adjective, to describe the low payments they may make every month for the things they buy. The military executive sometimes thinks of it as an appropriation asked for by the President and authorized by the Congress for the operation, maintenance and training of the naval establishment. All associate some type of book keeping with the word. To all visions of money are conjured. To many visions of products which can be bought with money immediately follows. To the man of the house the word is a symbol of the struggle to match income with outgo. To Mrs. Michael Joseph O'Shaughnessy the word became a veiled threat to her freedom and implied a vile lack of trust in her monetary judgement. In fact Elizabeth reacted violently each time Mike brought up the term in connection with their household activities.

Liz was the partner who managed the O'Shaughnessy household day by day. Her plan of action being routine and simple, any talk of a budget "to help save money" was so much nonsense. Only a few had a hand in expenditures. Their actions could be controlled in a most personalized manner. Emergencies happened. They were seen and felt by the entire household. Decisions were clearcut and obvious. She would buy hamburger for awhile or delay the purchase of a new dress she admired. The family checkbook provided all the essential elements required for the accounting system. A more complex system she viewed as being much ado about nothing. This was true through the years, and Captain O'Shaughnessy had to admit it.

But slowly the children grew older, and the family pattern changed. The simplicity of the small intimate group had been destroyed. Young Mike was at Dartmouth. Maureen had enrolled at University

* Navy Public Statements, July 1959.

of California. Mary was a senior at Trinity college in Washington. Liz was in the family home in Coronado and really had to stay there, when Mike got orders to an overseas position with the Seventh Fleet. Budget planning was now not merely attractive, but rather a necessity. The family goal remained as it was . . . the security, welfare and development of the members. But now each had his own independent and unique tasks which collectively would contribute to the O'Shaughnessy objective.

Captain O'Shaughnessy could follow either of two basic philosophies. Each is recognizable in our naval experiences to one degree or another. He could plan in detail every phase of the activities of each family member, thus assuring himself that while he was overseas, each would accomplish his or her task, and have the wherewithall to do it. After all he is an expert. He has been to college and has lived with the family since its inception . . . watched it grow, and trained each junior member. He is well aware of the weaknesses of Mike, and Maureen has a tendency to spend more time on boys than on books. He has the responsibility to pay the bills. He needn't ask anyone their needs, wants or desires. This might encourage them to be selfish and careless. Besides why burden them with the details of their requirements. He can very easily figure out what each needs for food, clothing, rent, tuition and the like. He knows all the answers. It will be simple to allocate his resources to the best interests of all. And best! How simple control will be. With this "standard plan," his accounting will allow him to catalogue each expense as it occurs, control quickly the individual who buys something "she doesn't need," and at the same time allow him to keep his income tax records in the proper order. By the simple expedient of giving authority to each member of the family to write checks on his checking account, and to annotate on each what was purchased, he will have a simple direct reporting system. Such a standardized system will allow positive control over the cash reserves of the O'Shaughnessy's! There can be no hidden costs! No initiative will be required. No one need worry but himself. How happy everyone will be for that! Success is assured. The system is alluring in its simplicity, and most appealing to the clearcut, orderly mind of Mike. It follows those principles for which he lives. Have an objective, formulate a clearcut plan of action, set up a system of evaluation from which quick, positive and effective decisions may be made!

But wait, Mike! What do Elizabeth and the children think about all this, for they are the ones who will be taking the actions to implement your program? What does Elizabeth have to say?

"Why you stubborn Irishman! What in the world do you think you're doing? Are you trying to promote cooperation or dissension in this family? Do you want your children to be loyal to you, or are you just trying to make them submissive. What are you going to do when Mary writes and says she needs money to pay a doctor bill? What's going to be your reaction when Maureen wants a new dress for the Junior Prom? Suppose that Mike decides he'd like to go to the Princeton game, and have a date in New York afterward? Will each be satisfied with your decisions then? If you have allocated all your resources what will you do? Will you threaten, plead or cajole? Will you demand economy and compliance, still knowing that your plan was good, and point out our responsibility to carry it out just as you outlined it? Will your family strive to save knowing that you will recapture what they do not spend? Or will they take the attitude, "the old man got us into this mess, let him get us out." The thing I fear most of all Mike, is that Maureen, Mary and young Mike aren't stupid. They'll figure out a way to beat your system if it means something which is important to them. You know the answers as well as I do, Mike! Your system is going to cause us all more trouble than it's worth."

"What's the alternative? We have a fine family. Let's get us all together during the summer. We all understand that you are responsible for our long range security. Mike, Maureen and Mary are mature enough to understand that you are responsible for my welfare, and that there are long years ahead for us and Katie. They will not want to jeopardize that objective. In fact you will bind us together in a common feeling, taking us all into your confidence, and letting us each do our little bit to help the family."

"Why don't you take these plans, as you have written them, and let each of us use them as a guide. Let's make it clear what the basic guidelines are. You've insured the family against catastrophe, but each of us is going to have minor crises which would only bother and worry you, if we kept sending you messages. Let's have each make up their own plan, which will take care of the year away . . . . . and let you take a look at it. If the total is more than we can afford, we can adjust here and there I am sure. You may even have to borrow a little this year, until Mary graduates. Make it clear to each of us that each will bear the full responsibility of living within the plans that each has made himself."

"The kids are children to us Mike! But they aren't really. It will be good for them to have the authority over their own expenditures, without too much interference. Mike and Maureen are going to have a little trouble at first, they're so happy go lucky. Mary will probably save and spend wisely right from the start. But in the long run we will all be happier for your faith and understanding."

"I know you still have that silly income tax form to make out! Oh, it really isn't silly, it's just that I can't make it out for you ever. I'm stupid I guess! But if you'll write out for each of us, the guidelines

of the information you need each month, I know the kids will cooperate. You could even make out some kind of a form, and we'll each put our cancelled checks with it . . . the ones that have to do with doctor bills, or contributions, or whatever else you need. That way you won't be bothered with a maze of detail . . . . . and you won't get mad at me every time I buy a new hat!"

Elizabeth was wiser than she knew. There still was an objective, the plan was as valid. Control of the basic elements of the program were really more rigid, because now no individual could hurt anyone but himself. Mike would still have his worries, but they wouldn't, God willing, be major calamities. Each decision would be aimed at a specific problem and at a specific individual. Common understanding and belief had been added. If centralized direction had been lessened, it had been strengthened by a faith that the actions of each individual would be more important. The fundamental dictates of human understanding had been added.

Yes, Liz was wiser than she knew. She would not understand completely why, but she would be pleased to know that the Comptrollership Foundation in a study of the best corporations in the United States revealed common characteristics which were very similar to her ideas. In a book* she could have read that these firms have:

1. A Budget system that provides for definite time schedules for the completion of formal plans, timely reporting of results related to objectives, and prompt control action related to organizational responsibilities.
2. A firm foundation for budget planning based upon sound organizational structure, which designates clear areas of authority and responsibility, coupled with an accounting system oriented to organizational responsibilities.
3. A formalization of the budget planning system and the control system, and the integration of these into a coordinated program consistent with the organizational responsibilities, and the planning and control needs.
4. Operating plans for the corporation and for each major organizational division thereof, formalized and expressed in detail.
5. A definite and clearcut assignment of responsibility for budget estimates related to operational assignments, and overall staff coordination of the process.
6. A system in which all levels of management actively participate in the planning and control process, and which recognizes the differences and the inter relationships between the two.
7. A reporting system which compares actual results with standards of objectives, by areas of responsibility, applying the exception principle.
8. A program of continuous budget education to insure the appreciation of the potentials involved, and active support and participation in the system from top to lowest levels of supervision.
9. A recognition of the importance of enlightened human relations in generating a spirit of willing cooperation among persons associated in organizations, and therefore a system designed to motivate people rather than to exert pressure on individuals.
10. A recognition of the fact that control is accomplished through people rather than through techniques, and that plans of themselves will not produce any action.

* "Business Budgeting" Comptrollership Foundation, Inc., 1958.

ACCOUNTING AND BUDGETING

This admittedly fictional story of the O'Shaughnessy family is realistic enough to focus our attentions on a few salient truths. A budget is in reality a plan of action, which if carried out will meet an objective. It describes on a time scale, actions of men in terms of manpower, facilities and materials, which will achieve a desired and specific result. Its chief utility as a management tool is that the resources which are employed or expended are reduced to a common unit of measurement . . . . money. Should we not have a monetary system in the United States, we would still be required to prepare some type of a budget, and the manager would search until he found a suitable common denominator. Perhaps units of energy could as easily be used to measure the value of the resources. Captain O'Shaughnessy was not talking about an expenditure of dollars to his family. He was speaking of the purchase of books, food, dresses and football tickets. Looked upon in this manner, a production plan is also a budget, for it too describes a series of actions in which manpower, materials and facilities are blended to achieve a desired product. The important thing is that the budget is a plan, and is useful because it serves as a standard against which actions may be evaluated to assist the naval executive in making decisions which will enhance his goal-seeking opportunities.

The associated tool required to assist the executive is an accounting system. Since dollars are a unit of measurement which can express the value of manpower, facilities or materials, we use them as a kind of shorthand. Using dollars to describe the actions which are planned to happen, dollars can also be used to measure that which did happen. Executives may use these building blocks to formulate a plan of action, store the information away in a ledger, or in an electronic data processing machine as a standard of what *should* happen. As actions occur, these may as readily be reported to the ledger, or to the electronic brain, as a description of what *did* happen. A comparison of the two will either demonstrate that actions happened as they were planned, or that there was a variation. In the latter case the executive must investigate and make a decision, unless he wants to allow his ship to go on the rocks. It may appear to the innocent that the machine is reporting only dollars. It is not. It is reporting actions . . . . . expenditures of resources.

In some instances accounting must serve a secondary process. In the case of Captain O'Shaughnessy he had a very real need to have some type of a system which would catalogue the expenditures of his family, because of the income tax laws. Within the Navy we have equally valid reasons for centralized accounting systems, which bear no relation to operations. Public Law requires that expenditures must be within the limits of that authorized by Congress for specific purposes. In addition

money authorized for specific purposes may not be spent for other items. Congress in other words has not authorized dollars, it has authorized manpower, or ships, or materials. In order to obey the law, accounting processes are required which will catalogue the expenditures of people, materials and facilities in terms of object classes, rather than operating actions. Travel by the day, or by the mile, is catalogued in terms of dollars, reporting to the Bureau of Naval Personnel in terms of dollars so that the law may be obeyed. Automobile maintenance is controlled by the Bureau of Yards and Docks. Materials, manpower and services expended for the maintenance of ships are segregated and reported to the Bureau of Ships. While this may seem unimportant to the Captain of a ship, it is not.

The real danger lies in a very real tendency to allow the secondary reason for accounting to overshadow, or even eliminate the primary reason, at the lower echelons. Within the Navy, a multiple accounting system is required, and does in fact exist. These tend to duplicate each other. They are expensive. The top level of Naval Management requires a standardized system which will produce the information required because of legislative requirements, the top level policies, or to assist in the development of long range programs. The lower level executive requires an accounting system directly geared to his individual type of operations. The operating executive may tend to allow his men to grow careless or indifferent to the centralized system allowing false data to flow through it. The top level may be indifferent to the requirements of the operating executive for his unique system.

The Navy Accounting System recognizes these facts. Naval officers sometimes do not, and hence do not use the tools available.

It is all well and good to speak of principle. It is easy to understand that in the extremes of the O'Shaughnessy family, such principles may be demonstrated, even though admittedly not practiced to perfection in *any* family including the Captain's. It may be just a coincidence that the study of the Comptrollership Foundation seemed to agree with the common sense approach of Elizabeth. Can we find an example in the Navy? Perhaps we may be able to describe many . . . . . certainly we need look no farther than the Bureau of Ships to find a near parallel.

BUDGETS FOR MAINTENANCE AND REPAIR OF SHIPS

The naval service creates an environment in which change is encouraged, and constructive criticism welcomed. That is our strength. The Bureau of Ships enjoys an enviable reputation of being a leader in the creation of such an attitude, not only among its fine corps of Engineering Officers, but in its solicitation of ideas from those who go to sea in command positions. None of us, who are honest with ourselves, will be able to quarrel with the slogan "Our only Mission is to Serve the Fleet." This is not to say that the Bureau of Ships is perfect, nor always has been. It is to say that, recognizing a lack of perfection, the Chiefs of the Bureau have with some consistency *sought* better ways of doing something. Equally important, having found it, they adopted it.

If you were the Commanding Officer of a ship in the late forties or early fifties, it may have happened that you attended an arrival conference at a Naval Shipyard and when it was finished you suffered some frustration. Your ship by policy was to have all repairs accomplished which were essential to her combat readiness. Your type commander had screened a work list and eliminated that work which was "non essential." Some preliminary planning had been accomplished on the work requests which had been transmitted to the shipyard. The objective was clear. The plan of action was equally clear. Yet at some point in the conference you would be shocked by the words "this is the cut off point on your work, Captain"; there is no more money to accomplish the remainder. If you later made loud enough representation to your type commander, a few more jobs may have been approved. As often as not, particularly if the month was May or June, your ship left the shipyard with some of the work not accomplished. The reason? The basic reason seemed to be that dollars appropriated for fleet maintenance did not meet demand for repairs. This of course was true. But why in your case did this come as a "surprise." This reason is more complex. Yet it is very similar to the reason why Captain O'Shaughnessy nearly ruined his family.

It was not difficult for the Bureau of Ships to forecast the costs of shipyard overhauls, based upon what costs had *been*. Using performance figures from Naval Shipyards, the accounting system then existent, and armed with the names of ships and overhaul dates supplied by the Type Commanders through the Chief of Naval Operations, the Bureau could predict, for instance, that the *average* destroyer overhaul would consume a specified number of manhours of direct labor, a predicted amount of material, and an overhead rate could be applied to the direct labor costs. Workloads could be spread as evenly as possible over the available shipyards, taking into consideration all the overhauls which were scheduled to be conducted during a given year. Funds could then be allocated, after the Congress acted, to the Fleet Commanders, who could suballocate to the Type Commanders. As an aid in allocation, and in order to check the distribution of funds, Type Commanders submitted a grand total figure of their estimated requirements, just before the fiscal year commenced. At first, this covered only a portion of the costs, "overhead and many materials being free." Later under the infant days of Industrial Funding, most of the costs were included.

Funds were allocated on a "return cost" basis; that is a type commander, except with a great deal of justification, could expect to get no more money, than he had spent the year before. Some officers in the type commanders' staffs, divided the total of the moneys allocated, by the overhauls for specific types, and established an average cost of overhaul figure. This was the amount made available for each ship of that type.

The system was direct and foolproof. It had the same flaws as Captain O'Shaughnessy's plan, and a few more when viewed against the standards of the best budget planning. Obviously the objective "to accomplish all repairs essential to the combat readiness of the fleet," was not assured by the plan. The plan being based on events which had happened in the past, and not those which were desired in the future. In a period of rising costs, the "return cost" concept aggravated this defect. Since the moneys were allocated in a lump sum, no time table of actions desired was available as a standard against which to measure actual performance. At the Bureau level of course there was. To the man in the staff there was not. Again there was no "average" ship being overhauled. By allocating a fixed sum to each overhaul, the Commanding Officer lost sight of the fact that essential repairs were his objective. He set out to spend a fixed amount of money, and usually did. The result was that the under-average ships in requirements somehow spent the money which should have been reserved for over average ships. The latter didn't get all essential repairs accomplished. One more factor aggravated the situation. The type commander sometimes had special programs which were inaugurated. These, being new items, would not appear in the historical statistics at the Bureau level. Lastly, since the Type Commander did not actually participate in the planning stage, and since he held a responsibility to insure that all ships were overhauled according to the Fleet Commander's policy, he tended at first to authorize complete overhauls, and as the year progressed tended to "run out of funds." His officers, in other words did not set up a plan, which if carried out would meet the objective for all ships, were therefore unable to compare what *did* happen, against what *should* happen, and certainly could not be guided into quick, effective decisions. There was no common doctrine throughout the echelons, incomplete centralized direction, and in this case uncontrolled decentralized execution.

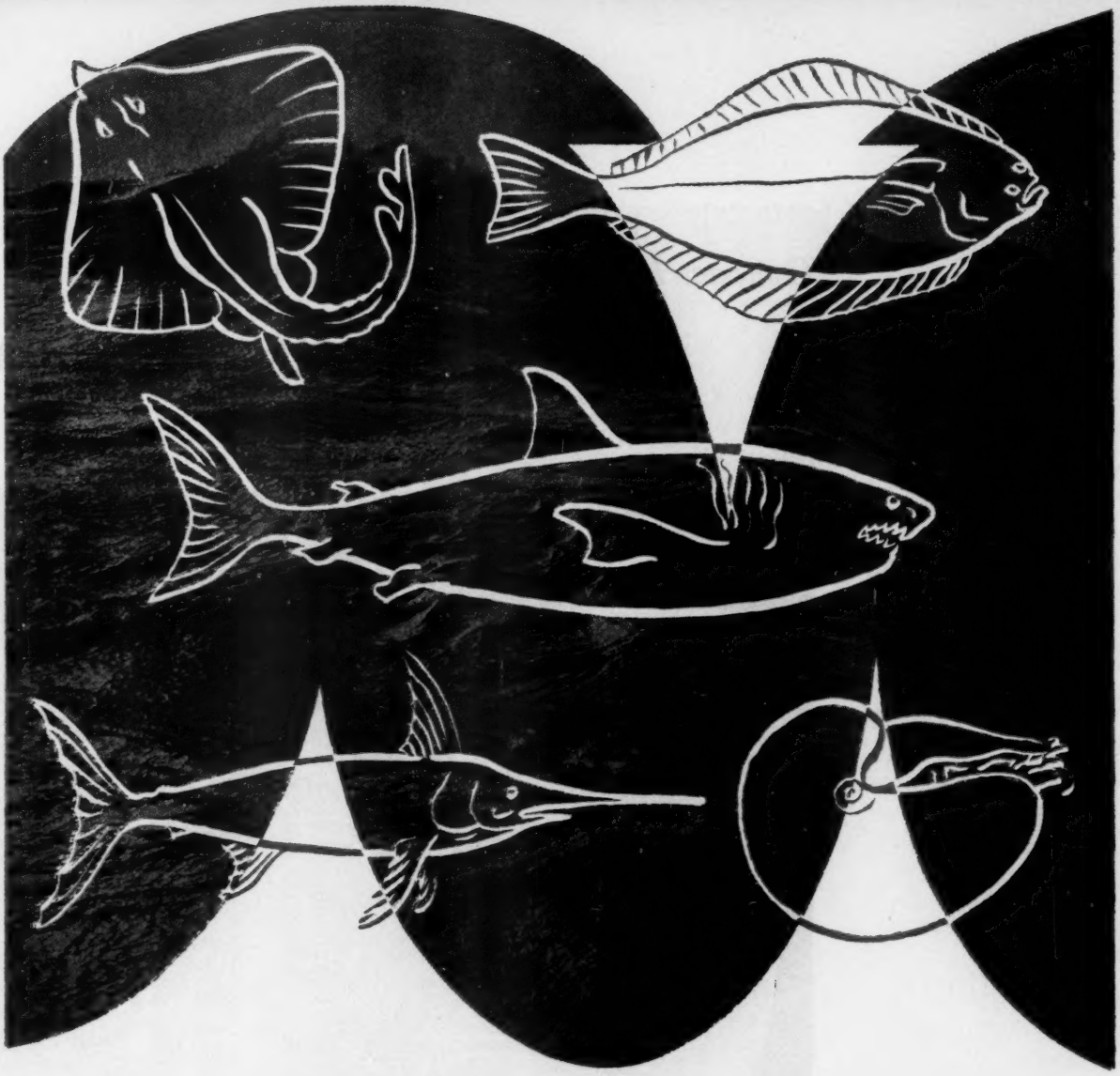
While this may sound rather obvious, in retrospect, at the time it was not. Naval Officers had not had to worry too much about money as such, in the preceding war years, and in these they did not fully correlate that dollars were rather meaningless, unless their minds focussed on the manpower and materials first, and then used money only as a unit of measurement. Demand *did* exceed supply, and in a period of rising costs and falling appropriations, there were more than a few frustrations facing the naval executive. Officers viewed accounting systems as "meaningless data" that the higher ups wanted, not as tools for comparison of planned and actual actions.

How did the Bureau solve the problem? Basically by adopting the same system that Elizabeth O'Shaughnessy described. By a steady program of indoctrination through conferences, directives, articles, and person to person education a common doctrine was established. Emphasis was placed upon the type commander making forecasts which were used in budget formulation, emphasizing forecasts of things to come, rather than "what had been." Emphasis was placed on objectives, and planning was based upon these objectives. Essential repairs, and the component resources, labor, overhead and materials were foremost in the naval executive's mind, rather than the dollars. Accounting took on an added importance with the advent of the electronic data processing machines. Industrial funding emphasized the importance of the customer in the planning process, and thus in the process of execution. Type commanders having prepared their plans, according to objectives, tended to compare their actions against their planned actions, and assumed more of a feeling of responsibility for these. If they wanted to add programs, after a plan had been formulated and money allocated, they had to give up something less essential, as a general rule. Decisions were more rapid, effective and realistic.

The Bureau had established a sound common doctrine, gave timely yet not restrictive direction, and facilitated decentralized execution.

In spite of continued rising costs, and emphasis on economy of operations, most commanding officers who are observant will notice a change in arrival conferences. I attended one last year in my ship. There was no "cut off line" below which my repairs would not be made. The emphasis was rather on "how essential is this repair?". The objective of the overhaul was met.

The Bureau of Ships system was developed by many officers, and refined year by year. Probably few, if any, had purposely tried to apply the essential ingredients of the best budget and accounting systems in the exact fashion as they are outlined in this article. Few had consciously thought about the Management Martini as such. None of course knew Captain Michael O'Shaughnessy and his titian haired wife. But they demonstrated in the actions the validity of these ideas, and the inevitable direction which Naval Executives will take when operating in the American Culture.



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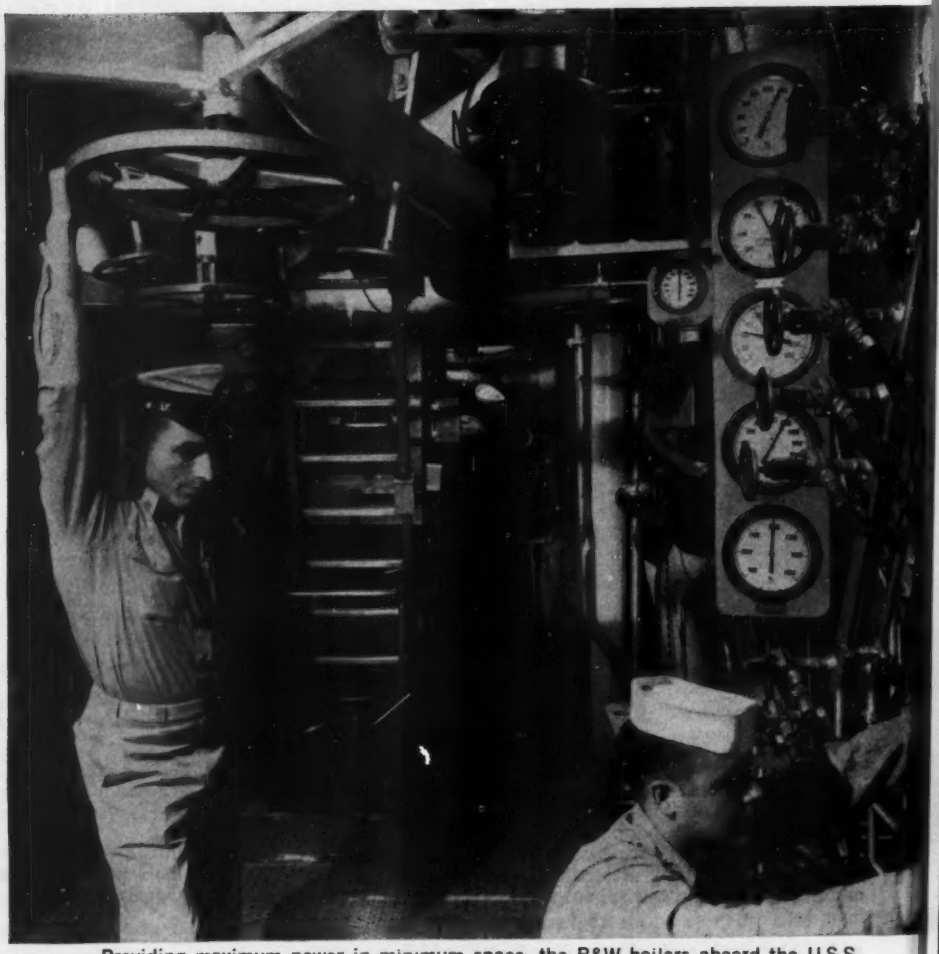

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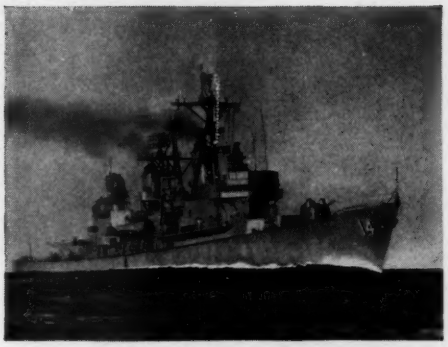

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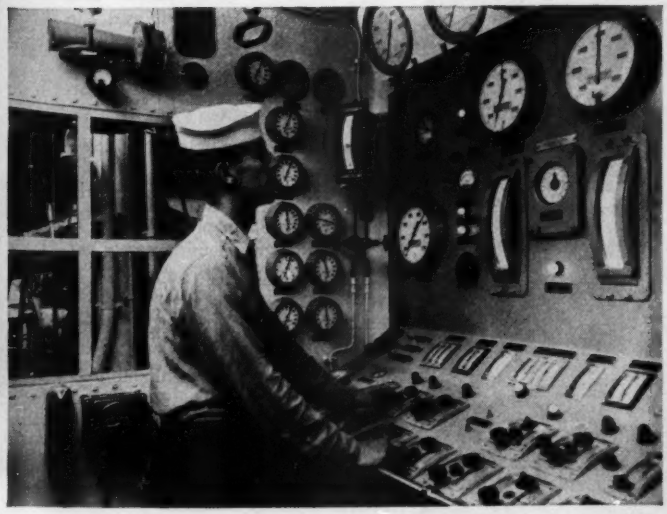

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GENERAL DEVELOPMENTS IN CERAMICS FOR MARINE ENGINEERING

ACKNOWLEDGMENT

This article, written by F. H. Aldred and N. W. Hinchliffe appeared in the March 1961 issue of "Transactions of Institute of Marine Engineers" from which it is reprinted.

INTRODUCTION

THE OXFORD ENGLISH Dictionary defines "ceramic" in the following terms:

> "Of or pertaining to pottery especially as an art; as a noun in the plural—the ceramic art."

It is quite obvious to anyone with even a superficial knowledge of ceramic technology that this definition is far too narrow to cover present day usage of the word.

There have in fact, in this century been several papers devoted entirely to attempts to define the term ceramic and one of them [11] concludes that Ceramics should be defined as an inorganic technology in which cold moulding is followed by consolidation by the action of heat. Although this has met with wide acceptance, in some directions it would seem to be too wide a definition, in that, for example, it could include sintered metals, but on the other hand it excludes hot-pressing which is now an established ceramic technique.

The American Ceramic Society includes among its interests; pottery, clay building materials, refractories, hydraulic cement, vitreous enamel ware, glass, abrasives and electro ceramics. The interests of the British Ceramic Society are not so wide, in that it has four sections; Pottery, Heavy Clay Ware, Refractories, and Basic Science. The first of these sections embraces industrial porcelains and electrical ceramics as well as true pottery.

Taking even the more restricted British interpretation of ceramics these are involved in the marine industry in many ways such as:

(a) In the metallurgical industries, which represent the biggest tonnage consumption of refractories.

(b) In the fabrication of metals, e.g. refractories in furnaces, ceramic welding nozzles, ceramic sand blast nozzles.

(c) In heat engines, not only in traditional boiler plants but in gas turbines and nuclear power plants.

(d) In electronic equipment, which includes a wide range of ceramic materials chosen usually for specific electrical properties.

(e) Lastly, ceramics of course are used in the

"domestic" sense in sanitary ware, tiles, and pottery.

It is proposed to limit the present paper to the use of ceramic materials in heat engines, and to cover not only established uses but also work which is still in the very early development stages and may or may not ultimately have commercial impact on marine engineering.

MARINE BOILER LININGS

Firebricks

At the end of the last war it was apparent that the refractories available in Great Britain for lining marine boilers were inadequate to meet the increasingly severe operating conditions in boilers of new design. The choice of materials was almost entirely confined to firebricks made from Scottish clays, having analyses in the range 35 per cent Al_2O_3, per cent SiO_2 to 42 per cent Al_2O_3, 54 per cent SiO_2, the balance in each case being made up of fluxes such as mica, felspar and the oxides of iron and titanium.

These grades were kilned to temperatures of the order of 1,350—1,400 deg. C. resulting in limiting service temperatures up to about 1,450 deg. C. in the case of the 42 per cent alumina grade and 1,350—1,400 deg. C. in the case of the 35 per cent alumina grade. The majority of these bricks were moulded by either soft mud or stiff mud processes, which, because of high drying and kilning shrinkages, leave much to be desired in shape and dimensional accuracy.

Improved service results were obtained by the introduction of dry and semi-dry moulding techniques which produce bricks of a much improved dimensional standard. Troubles still arose however, from bricks, especially quarl brick, spalling after quite short periods of steaming and from slagging which generally attacked the joints to a greater degree than the remainder of the lining. This joint attack was aggravated by brick shrinkage in parts of the boiler where temperatures [14] were near the safe limit for the refractory and by the use of clay jointing materials which were inadequate for the purpose.

It was apparent that there was a need for refractory bricks which had even better thermal shock resistance and greater volume stability than the best quality Scottish firebricks and development of suitable products was undertaken by several manufacturers.

It appeared desirable to develop bricks based on indigenous raw materials and here the first problem was to find a suitable clay which would withstand the higher kilning temperatures required to give adequate volume stability. Two possibilities were open, either the benefication of the 42 per cent Al_2O_3 Ayrshire clays in order to reduce the flux content or the use of china clays which have low flux contents and are generally more refractory.

The manufacture of bricks from china clays represented quite a new departure and one of the major problems facing the manufacturers was the very high linear shrinkage associated with the kilning of these clays when temperatures of the order of 1,600 deg. C. are used to confer stability. This was overcome by employing as the main constituent of the refractory, china clay which had already been calcined to a sufficiently high temperature virtually to eliminate shrinkage. This calcined material, usually termed "grog," formed up to 80 per cent of the final product. By careful size grading of this grog, bricks were produced which on laboratory test had very much better thermal shock resistance than the best quality Scottish firebrick. These results have been fully confirmed in service.

At the same time, development proceeded with refractories based on imported raw materials. Sillimanite ($Al_2O_3 \cdot SiO_2$) and bauxite were employed, the development following a similar pattern to that with china clays, and products were evolved ranging from 55-65 per cent alumina content, having excellent volume stability at temperatures well in excess of 1,500 deg. C. These products also showed excellent resistance to thermal shock.

Typical properties of the bricks which have resulted from this development work and which are now in service in marine boilers, are shown in Table I, compared with a typical Scottish firebrick.

Almost all the china clay and high alumina refractories are manufactured by dry pressing methods ensuring consistency and accuracy of size. Air setting cements especially developed for use with these grades are recommended for use in marine boilers, and eliminate the jointing troubles referred to earlier.

TABLE I—*Properties of Bricks for Marine Boiler Linings*

	Scottish firebrick	China clay Super Duty	Bauxite based 60 per cent Al_2O_3	Sillimanite based 55 per cent Al_2O_3	Sillimanite based 63 per cent Al_2O_3
Per cent Al_2O_3	42	43	60	56	63
Per cent SiO_2	51	53	36	39	34
Per cent total flux content	7	4	4	5	3
Bulk density gm/cc.	1.92	2.15	2.15	2.31	2.35
Apparent porosity, per cent	28	17	23	25	22
Reheat shrinkage 2 hrs. at.	0.3 per cent (1,410 deg. C.)	0.8 per cent (1,500 deg. C.)	0.8 per cent (1,600 deg. C.)	+0.5 per cent (1,500 deg. C.)	0.2 per cent (1,500 deg. C.)

Thermal Shock Testing

In all developments it is necessary to have some means of assessing the product in the laboratory, and while reasonable test procedures existed for determining physical properties, the only test for thermal shock resistance in use in this country consisted of heating a 2-in. prism to 1,000 deg. C. repeatedly until it fractured. This test has obvious shortcomings especially for applications where it was anticipated that there would be rapid temperature fluctuations up to 1,400 deg. C. In the United States, the American Society for Testing Materials developed a panel spalling test which tests a panel of bricks built in a similar manner to normal construction and tested by heating on one face as would be the case in service [1 and 15].

The panel spalling test for Super Duty bricks is a two part test comprising a preheat test in which the face of the panel is heated to 1,650 deg. C. for 24 hours to simulate the effect of prolonged heating in service followed by a spalling test in which the face of the panel is heated to 1,400 deg. C. for 10 minutes and cooled by means of a controlled air and water blast for a further 10 minutes. The heating and cooling cycle is repeated 12 times and the loss in weight of the test bricks at the end of the test is taken as a measure of the resistance to spalling.

Figure 1, illustrates a panel of 42 per cent Al_2O_3 Scottish firebricks which lost 9 per cent by weight, after testing on the A.S.T.M. panel spalling test.

Figure 2 shows a panel of china clay Super Duty firebrick which lost 3 per cent by weight on test, and Figure 3, a 60 per cent alumina bauxite based high alumina brick which showed no loss on test.

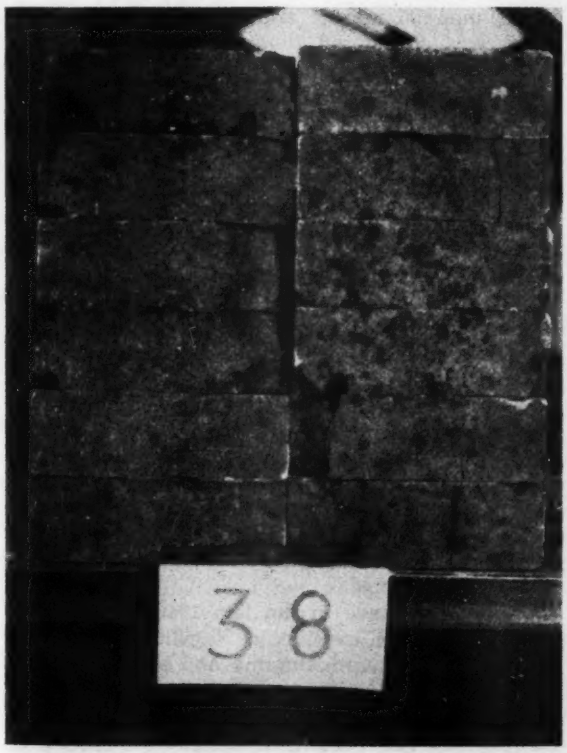

Figure 1. Scottish firebrick after panel spalling test.

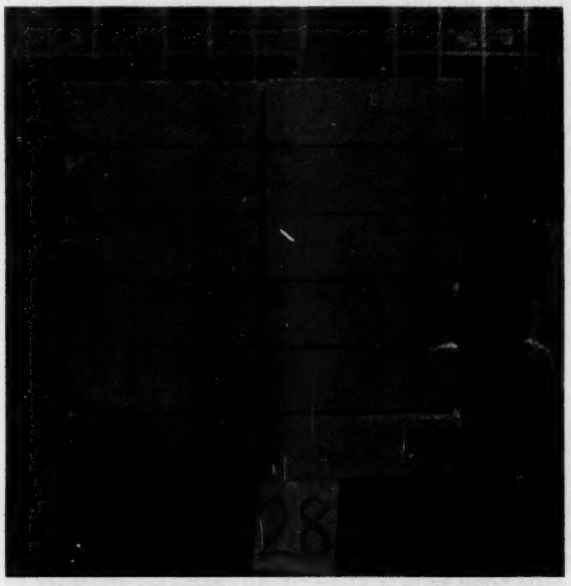

Figure 2. China clay brick after panel spalling test.

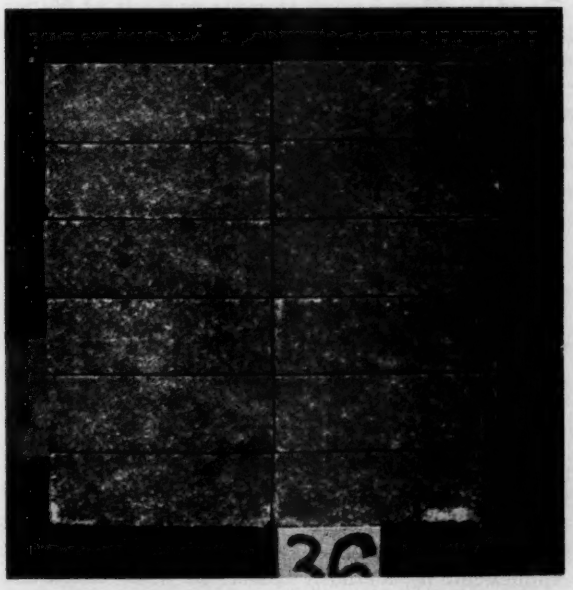

Figure 3. High alumina brick after panel spalling test.

Insulation

Insulating firebricks capable of withstanding temperatures in excess of 1,500 deg. C. and having thermal conductivities of about one third that of normal firebrick, have also been developed. These are based on various clays which in manufacture are blended with a combustible material such as sawdust or coke. The combustible burns out during kilning leaving a very porous light-weight refractory. These products permit higher interface temperatures between the solid grade lining and the insulation so that either thinner hot face linings or a greater amount of insulation can be employed.

Monolithic Refractories

The other major refractory developments affecting marine boilers have been in the field of mouldable and castable refractories. In the furnace chamber the use of mouldables has become widespread and for lining superheater chambers, castables have been used with very satisfactory results.

Pre-1950 mouldable refractories were simply firebrick recipes supplied in the unfired state and used for patching spalled or eroded brickwork. These patches frequently gave very good results in service and so steps were taken to modify the recipes so that the products could be used for massive installations. The important step in the development was to increase the grog content of the mix substantially which necessitated replacing the fireclay bond with a more plastic ball clay in order to retain good moulding properties. Higher temperature plastics were developed based on the same raw materials as the aforementioned bricks. Those having 40-45 per cent Al_2O_3 are usually suitable for service temperatures up to 1,450 deg. C.—1,500 deg. C., while others having alumina contents of 60 per cent or more are suitable for temperatures of the order of 1,600—1,650 deg. C.

Mouldable refractories have better resistance to thermal shock than can be obtained in almost any prefired refractory. Although prefiring a mouldable refractory reduces its resistance to thermal shock in proportion to the degree of vitrification of the bond, the thermal gradient through a structure ensures that a large part of the refractory retains this excellent resistance.

At first, mouldables were used only for forming the burner quarls but their use spread rapidly to the whole of the front wall and in many boilers, to side and rear walls as well. As the use of mouldables extended it became necessary to devise methods of providing effective anchoring throughout the lining thickness while retaining free movement between the furnace casing and the lining. Figure 4 illustrates a method which has proved effective for walls of varying thickness.

Refractory castables have also been developed for application by methods similar to those employed with structural concretes. These materials have

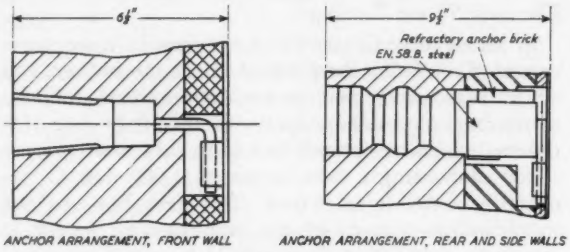

Figure 4.

good resistance to thermal shock but are less flexible than mouldables. They generally find use in the superheater sections of boilers where the freedom from joints ensures that hot gases are kept away from the casing. Anchoring for the castables can usually be effected with metal anchors and where temperatures on the anchoring do not exceed 300-350 deg. C. mild steel expanded metal can be used safely. Where temperatures are higher, it is necessary to use heat resisting studs which should be wrapped with paper before placing the castable. The paper will char away leaving space for the greater expansion of the metal than that of the castable.

THE MARINE GAS TURBINE COMBUSTION CHAMBER

In the immediate post war years, intensive research was directed towards developing gas turbines for commercial marine application. From the outset it was considered that the metal combustion chamber, as used in aircraft type gas turbines would corrode too rapidly to permit economic marine application particularly if heavy grade fuel oil was used. It was necessary therefore to devise some means for protecting the metal by a refractory material. The use of such a lining would have the added advantage of increasing radiation so improving combustion conditions.

The requirements of the refractory were:

1. The ability to withstand temperatures up to 1,750 deg. C. for prolonged campaigns.
2. The ability to withstand severe heating and cooling shocks.
3. Strength enough to withstand mechanical vibration.
4. The ability to withstand the action of oil fuel slags without spalling or peeling.

All these factors were considered to be of extreme importance because any breaking up of the refractory could prove costly in damage to the turbine rotor and blades.

Trials were made in the late 1940's using three materials, silicon carbide, a 70 per cent alumina body based on fused alumina and an 80 per cent alumina body kilned to both intermediate and very high temperatures. The trial linings consisted of tongued and grooved segments approximately 6 in. × 6 in. × 3 in. thick.

These trials indicated that the 70 per cent alumina body was insufficiently refractory to withstand long operating campaigns. The silicon carbide oxidized and broke up and the high temperature fired 80 per cent alumina body spalled. The intermediate temperature fired 80 per cent alumina body showed the most promise. In order further to improve the shock resistance of the lining, it was decided to reduce the thickness of the refractory to about 1 in. and to suspend each tile on lugs welded to the combustion chamber shell.

The modifications proved effective and a great many test starts and shut-downs were carried out with only very minor cracking. A few very small pieces did in fact, break off the refractories during trials but no damage to the turbine resulted and it was concluded that gas velocities were not sufficiently high to carry pieces of refractory large enough to damage the blades.

Since these trials, several gas turbine combustion chambers have been lined successfully with this type of refractory although only short periods of their service have been on heavy grade fuels [12].

There are still many problems to be overcome in combustion chamber linings, particularly as the addition of fuel oil additives to reduce blade corrosion has introduced other complications which may be more difficult to overcome than the earlier problems. Magnesium salts have proved to be very effective in reducing corrosion but at temperatures above 1,250 deg. C. these severely flux aluminosilicate refractories. In one case known to the authors, the refractory lining was almost completely eroded away during 100 hours operation. The final solution may be in the use of magnesia-alumina spinels but very much work remains to be done in order to produce refractories from such a body that will meet the other requirements of thermal shock and mechanical strength.

GAS TURBINE BLADES

General

The fuel economy of the gas turbine, as is well known, improves as the gas inlet temperature to the turbine increases. It has, for example, been stated that a temperature of 1,200 deg. C. is required to achieve a fuel economy comparable with that of a marine Diesel engine [5].

It seems unlikely that metals can operate at this temperature for long periods such as those required of a marine engine unless some technique of cooling is used, so that for some years there has been an incentive to develop other materials for the purpose.

Ceramic materials naturally come to mind in this connection and certain of them have the necessary resistance to temperature and corrosion in good measure. However, they are brittle materials readily damaged by mechanical shock and by rapid temperature changes (thermal shock), and are, in general, unreliable when subjected to tensile stress.

At the end of the last war a considerable amount of work was initiated to develop improved ceramics for use as gas turbine blade materials. After the first flush of enthusiasm it was realized that there was little hope of using these materials for rotor blades so that attention was concentrated on the still difficult problem of application to stator blades.

As far as the authors are aware there is no non-metallic material at present in use, even for stator blades, but there appear to be at least two special ceramics of the non-oxide type, silicon carbide and silicon nitride, which come near to fulfilling the requirements for stator blades in a high temperature gas turbine and which might, with further development, be applied in practice.

Some of the research which has been reported is described here, exemplifying the search for improved ductility (while retaining good creep strength) high thermal conductivity and low thermal expansion, all of which associated with high breaking strain lead to an improvement in thermal shock resistance.

Metal-ceramic Compositions

Following early work in Germany towards the end of the last war, a considerable amount of research has been carried out in many countries in an attempt to combine a metal and a ceramic so that the final composition has the ductility of the metal combined with the high temperature strength and creep resistance of the ceramic.

It has not in the event proved possible to combine the best properties of the two components. In general, starting from a metal and adding a ceramic material in increasing proportion, where the ceramic particles are relatively large there is at first little effect on the properties of the metal. There may be some improvement in creep resistance in special cases, where the ceramic is finely dispersed. With increasing addition of the ceramic material, there is transition from a continuous metal phase to a continuous ceramic phase, and the composition then has properties little different from those of the ceramic. This is well exemplified by the effect on electrical resistivity shown in Figure 5. Much ingenuity has been expended to produce metal-ceramics in which the properties of both phases are retained but little practical success has been achieved. The research has however led almost directly to new techniques used to manufacture the special ceramics referred to below.

A test, in a gas turbine test rig, of one of these metal-ceramic compositions is reported by Blakeley and Darling [5].

The resistance to thermal shock appeared quite promising, but this had obviously been obtained at the expense of the high temperature properties of the ceramic phase, since the blades were reported to be weak in creep and distorted appreciably in 2 or 3 hours testing at 1,200 deg. C.

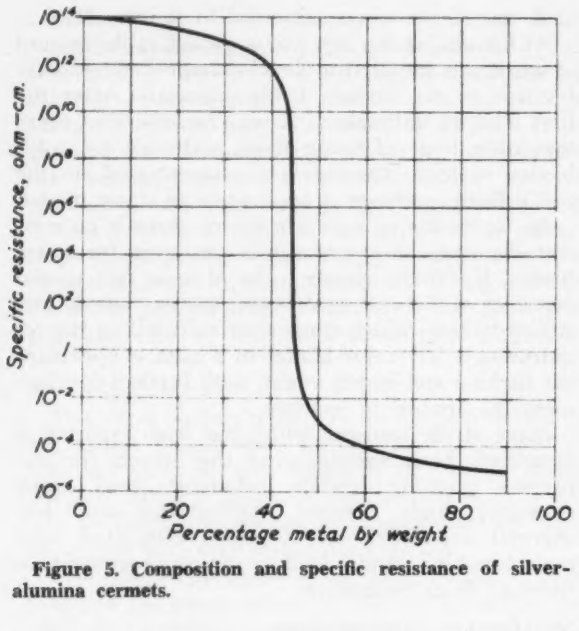

Figure 5. Composition and specific resistance of silver-alumina cermets.

Ceramics of High Thermal Conductivity

Another class of ceramic considered for turbine blades includes those of high thermal conductivity. For example at 1,000 deg. C. beryllia has a conductivity of 0.046, and silicon carbide 0.072 compared with 0.014 cals/cm. sec. deg. C. for alumina. These two former values are as high as those of many metals at high temperature.

Beryllia is extremely toxic and it may be for this reason that no work appears to have been done on it for this application. Silicon carbide in many ways is an excellent ceramic material, though limited in application by its proneness to oxidation and its reluctance to sinter. Of recent years much research is known to have been done on it to improve both these properties, but little has been published. Some work by Popper at the British Ceramic Research Association is an exception [17].

A set of stator blades in one of the commercially developed materials ("IC" silicon carbide) has been tested in a gas turbine test rig and showed promise of running for thousands of hours without cracking or distortion [5]. Further work [6] has confirmed this promise and overcome some of the fabrication difficulties referred to in the 1957 paper.

Low Expansion Ceramics

Fused silica is well known to have excellent thermal shock resistance, mainly by reason of its low coefficient of thermal expansion (0.5×10^{-6}/deg. C.). Unfortunately, this material has poor creep resistance and with prolonged heating at temperatures approaching 1,200 deg. C., devitrifies and loses its thermal shock resistance. It is equally true that other low expansion oxide ceramics such as the lithium silicates and cordierite (magnesium aluminum silicate) have relatively poor creep resistance. The search for other materials of low thermal expansion has led to an investigation of non-oxide materials, particularly silicon nitride [16].

Silicon nitride is manufactured by a new technique known as reaction sintering. A compact of silicon powder is first made, best results being obtained by isostatic pressing. This is then heat-treated in a nitrogen atmosphere so that the compact is converted to silicon nitride and simultaneously sintered. In practice this nitriding is carried out in two stages which has the added advantage that the intermediate material is easily machinable. Since there is only very slight shrinkage in the final heat treatment, articles can readily be made to close tolerances. This eliminates one of the problems of conventional ceramic manufacture where the high shrinkage on sintering renders dimensional control difficult.

The thermal shock resistance of silicon nitride is reported to be excellent and approaching that of metals, and it is dimensionally stable and resistant to oxidation up to 1,400 deg. C. The creep resistance of "pure" silicon nitride is not as good as that of some oxide ceramics, but it can be improved by incorporating a proportion of silicon carbide into the structure.

DIESEL ENGINES

General

The brittleness and poor resistance to tensile stresses typical of ceramic materials do not commend these to the designer of reciprocating machinery such as Diesel engines. Against this the excellent abrasion resistance of ceramics is attractive, and there have been some attempts to use them in, for example, fuel injector nozzles. However, the good abrasion resistance also implies that these materials are difficult and expensive to machine; in particular the machining of fine holes to close tolerances presents serious difficulty, and it may be this which is the limiting factor in this connection.

Attempts have also been made to use ceramic coatings to protect metal components from the effects of high temperature corrosive gases, initially with limited success, but new techniques appear to have given fresh impetus to this work.

Ceramic Coatings

Early attempts to use ceramic coatings were based on enamelling techniques, and achieved only limited success. Broadly speaking, an enamel is a ceramic type composition of fairly low melting point and usually of a glassy nature (known as "frit"). This is finely ground, suspended in a liquid and applied to the metal. Subsequently the coated metal article is baked or stoved when the liquid volatilizes and the enamel frits together, and also

bonds to the metal. Many refinements are of course involved, in the form of multiple coatings and pretreatment of the metal.

One limitation of the process is immediately apparent from the above description; the metal has to be heated to a temperature at which the coating will frit or sinter. Thus, if the metal is to be enabled to work at a temperature above its normal working limit, the coating must be designed to frit at a temperature below this limit and yet be serviceable at a higher temperature. This difficult, though not impossible feat, severely limits the range of ceramic compositions which can be employed and usually one advantage of the true enamel is lost in that the coating is not impervious to gases and liquids. Thus, at best, it acts as a thermal barrier only, reducing the surface temperature of the metal.

Recently, techniques known collectively as flame spraying have been introduced and alter this situation to some extent. In flame spraying, ceramic material in the form of rod or powder, is fed through a high temperature flame and a blast of hot gas directs the material in the form of molten droplets on to the surface to be coated, which is not heated to any great extent. Again, however, the coatings are not completely impervious but it has been claimed [8] that they can be made so by the use of sub-coatings or by impregnation.

These coatings have been proposed [8] for:

1. Inhibition of oxidation.
2. Protection against wear.
3. Control of heat radiation.
4. Fireproofing.
5. Electrical insulation.
6. Thermal insulation.
7. Protection against corrosion.

In particular the application of flame sprayed coatings to Diesel engine pistons and valves has been reported [2 and 4]. The ceramic materials used were the oxides of rare earth metals such as cerium, lanthanum praseodynium and neodynium, and these are reported to localize heat in the combustion chamber and also to have a catalytic effect ensuring more complete combustion and a cleaner engine. Claims are also made that oil changes are less frequent and that the decreased content of carbon monoxide and unburned hydrocarbons in the exhaust gases contributes to the reduction in atmospheric pollution at present being sought in most civilized countries. The effect of cheaper oxide coatings, such as alumina, does not appear to have been reported upon.

NUCLEAR PROPULSION

General

One of the more recent technological developments of potential interest to the marine engineering industry is the generation of power by nuclear fission and its application to ship propulsion.

As is now well known the nuclei of certain heavy atoms when bombarded by neutrons split into roughly equal parts; a considerable amount of energy is released and several neutrons are ejected. These are also capable of producing fission in other heavy nuclei so that under suitable conditions a self-perpetuating chain reaction can be formed. The fissile materials are U^{235} which occurs naturally as 0.7 per cent of natural uranium, U^{233} which is formed by neutron absorption in thorium and Pu^{239} which is derived from uranium. Unlike the fossil fuels the release of nuclear energy does not require air for combustion and the energy potentially available in one pound of U^{235} is equal to that obtainable by burning 1,500 tons of high grade coal. The attractions for ship propulsion, particularly in the case of submarines, are obvious.

Since the nuclear energy is released initially in the form of heat, and this has to be converted into other forms of energy, the overall efficiency of nuclear power generation increases as temperatures are raised. There is thus a demand for materials which will have appropriate nuclear properties and which will be resistant to corrosion by specific gases and liquids used for heat transfer, while also withstanding relatively high temperatures.

The thoughts of designers again turn towards ceramics and much research and development has been done to produce materials suitable for specific application in nuclear reactors, with such success that these materials are already in use in some installations.

Nuclear Fuels

The obvious material for use as a nuclear fuel is natural uranium, but this has a number of disadvantages [7]. Briefly, because of phase changes and anisotropy, the metal distorts on thermal cycling and in addition its maximum working temperature is limited to approximately 450 deg. C. by swelling under irradiation. Attempts to produce alloys free from these effects have been partially successful, but there appears to be a trend towards ceramic fuel both for high and low temperature systems.

Uranium has four oxides; UO, UO_2, U_3O_8, UO_3 of which UO_2 is the most stable and has a melting point of approximately 2,800 deg. C. compared with 1,130 deg. C. for the metal. This oxide shows a range of non-stoicheiometry ($UO_{2.0}$—$UO_{2.2}$)* and on being heated in air at 450—500 deg. C. forms U_3O_8 with an increase in volume [13]. These effects have a profound influence on the fabrication of uranium dioxide and much work has been done to obtain a material having the high density (95 per cent of theoretical) desirable for nuclear reactor use.

Fabrication must be carried out in a non-oxidizing atmosphere and higher densities can be obtained with non-stoicheiometric oxide than with the

*i.e. can contain oxygen in excess of the theoretical amount.

TABLE II—*Properties of Various Nuclear Fuels* [7, 13, 18]

Property	Uranium	Uranium Dioxide	Uranium Monocarbide
Coefficient of thermal expansion (10^{-6}/deg. C.)	(a)† 20.3 (b)† — 1.4 (c)† 22.2	10.0	11.4
Thermal conductivity (cal./sec. cm. deg. C.)	0.071	0.017	0.060
Ultimate tensile strength (lbs./sq. in.)	approximate 100,000	—	—
Bend strength (lbs./sq. in.)	—	16,000	50,000

† Crystalline axes.

stoicheiometric. For example, a particular oxide of composition $UO_{2.03}$ gave a density of approximately 7.1 gms/cc when sintered to 1,400 deg. C. but this was increased to 8.75 gms/cc when the powder was oxidized to $UO_{2.17}$ [13]. As is usual in ceramic technology, high surface area also favors sintering to high density.

Many ceramic oxides can be sintered to high densities by "hot pressing," a technique in which pressure and temperature are applied simultaneously in graphite dies. In the case of uranium dioxide chemical reduction of the surface of the compact takes place giving differential shrinkage and cracking [13].

A disadvantage of uranium dioxide as a nuclear fuel is its low thermal conductivity, which leads to a high temperature at the centre of the element. Interest has therefore been aroused in the uranium carbides, UC and UC_2, particularly the former, since the dicarbide tends to be unstable in moist atmospheres. Some properties of uranium monocarbide compared with uranium and uranium dioxide are shown in Table 2.

Uranium monocarbide is pyrophoric so that considerable precautions are necessary in handling fine powders, quite apart from radiation hazards. It is difficult to fabricate into a dense material of fine grain size and good mechanical properties. Arc melting of a mixture of uranium and carbon followed by casting has been used to prepare UC, in addition to the more usual techniques of compacting and sintering [18].

Another ceramic material which can be classed as a nuclear fuel is thorium oxide. Thorium absorbs a neutron and, by a process of radioactive decay forms U^{233} which is fissile. Thoria is one of the highest melting oxides (3,200 deg. C.) and is difficult to sinter to high density. It has been found that the addition of lime increases the density on firing at 1,700 deg. C. from 86 per cent to 96 per cent of the theoretical density [10] while similar densities are now being achieved commercially by the authors' company without additions, the resulting material analysing 99.9 per cent ThO_2.

Many other ceramic materials have been suggested for use as nuclear fuel and brief reports on them have appeared. Examples are an Fe - UO_2 composition [13] and a plutonium oxide—uranium oxide mixture [7] which would make use of the large quantities of plutonium which should shortly become available from the operation of land based nuclear power stations. Undoubtedly much work is in hand in many laboratories to provide materials for high temperature fuel elements, and it appears that much of this work has a ceramic flavour.

Thermal Barrier Discs

Current reactor techniques involve encasing the fuel element in a can, which separates the fuel element from the coolant to prevent corrosion of the former and contamination of the latter. Additionally the can acts as a structural unit. Once again there is an obvious trend towards higher can temperatures.

The use of ceramics as canning materials would involve considerable technological advances since the can must be reliably impervious in long periods of operation, during which thermal and possibly mechanical shocks could occur. Since most ceramics are of relatively low thermal conductivity heat transfer problems would also arise.

TABLE III—*Properties of Oxide Ceramics* [19]

Material	Neutron absorption cross-section (cm.²/cc.)	Coefficient of Thermal conductivity at 100 deg. C. (cals./sec. cm. deg. C.)
Beryllia	0.00074	0.500
Alumina	0.0101	0.069
Magnesia	0.0032	0.082
Zirconia	0.0057	0.004
Hafnia	3.18	—

However, ceramic materials are already being applied in connection with fuel element cans, in one of their traditional roles as thermal insulators. If the uranium metal fuel element is in direct contact with the can the contact temperature is limited by metallurgical considerations. Consequently in some designs ceramic discs are interposed between the element and the can to act as a thermal barrier. The ceramic must have low neutron absorption and

good long term stability under irradiation, in addition to as low a thermal conductivity as possible. The relevant properties for a number of materials are shown in Table 3.

Of these materials, beryllia, although having a low absorption cross-section, has an exceptionally high thermal conductivity for an oxide while zirconia would seem to have the best combination of thermal conductivity and neutron absorption. The usual commercially obtainable zirconia however contains approximately 2 per cent of hafnia which markedly increases its neutron absorption. While separation of the hafnia is possible it is a costly procedure and this, coupled with some doubts about the irradiation stability of zirconia, has kept this material in the background as a thermal barrier ceramic. In fact little development appears to have been done on materials for this comparatively minor component of the reactor, and established dense high alumina, or less commonly magnesia, ceramics have been used.

There would appear to be scope for investigation of the long term stability of other low conductivity ceramics, such as those having porous structures or those having intrinsically low conductivity. One example of the latter class is forsterite ($2\ MgO.SiO_2$) which the authors have prepared in high purity form with a thermal conductivity of 0.008 cal/sec. cm. deg. C. at 100 deg. C. and a measured neutron absorption cross-section of 0.0036 cm^2/cc. In view of its purity this material may have better long term stability than the conventional forsterite ceramics. Techniques for controlling the porosity of alumina and magnesia ceramics are also known, and enable materials of somewhat reduced thermal conductivity to be manufactured.

Control Rods

A further example of research in the ceramic field designed to meet the requirements of the nuclear reactor is provided by the problem of controlling the reactor. This is carried out by adjusting the position of rods of neutron absorbing material in the reactor core, the material in common use being boron, in the form of boron steel. Some other elements having a high neutron absorption cross-section are shown in Table 4.

TABLE IV

Thermal Neutron Absorption Cross-Sections

Element	Cm.²/gms.
Cadmium	12.8
Europium	17.9
Samarium	26.0
Boron	42.0
Gadolinium	169.0

It is not possible to incorporate a very high percentage of boron in steel so that control rods of boron steel are heavy and necessitate expensive and bulky electrical control gear for their operation. Additionally natural boron breaks down under irradiation with release of helium, which at the higher temperatures of operation can cause swelling of the control rod.

For these reasons there is interest in the use of the three rare earth metals, samarium, gadolinium and europium in various forms including oxide ceramics. It is believed that these materials will enable much lighter control rods to be made, and that these will be stable under irradiation.

The three oxides, samaria, europia and gadolinia have high melting points, and initial research showed that they are comparatively easily fabricated into low porosity ceramics. However, the geometrical requirements of the control rod are such that these simple materials would require to be fabricated into impossibly thin sections to avoid the system having too high an absorption. Research has therefore continued on mixtures of the rare earth oxides with readily available low absorption ceramics so that the absorbing nuclei are distributed over a larger volume. This has necessitated some quite fundamental research and, for example, phase diagrams have been determined [3]. A typical example is shown in Figure 6; the other combinations of rare earth oxides and common ceramic oxides give similar diagrams. If compositions near the eutectic are required severe ceramic problems arise, which are intensified if unseparated mixtures

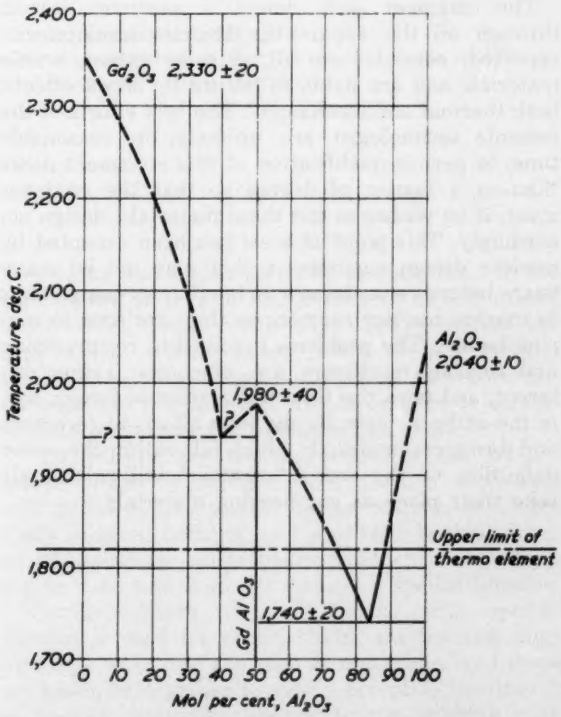

Figure 6. Tentative diagram for the system Gd_2O_3—Al_2O_3.

of rare earth oxides are used for economic reasons.

Other materials which have been developed in the search for new and improved ceramic materials may also be of value in controlling nuclear reactors. For example, boron nitride has been described [9] and boron carbide is commercially available.

Of just as much importance as the initial neutron absorption cross-section of the control rod is its total capacity for absorbing neutrons and the changes which take place on irradiation, since these will determine its life. This necessitates a knowledge of the properties of the daughter products of the absorption process, and this type of work is being undertaken by nuclear physicists to enable the choice of the most suitable control rod materials to be made. If these materials prove to be "ceramic" the problems involved in the use of brittle materials of limited thermal shock resistance will need consideration.

CONCLUDING REMARKS

It has not been possible in the space of this survey to deal in detail with the many ceramic developments which may be of significance to the marine engineer. It is hoped that sufficient has been said to indicate the enormous increase in the scope of ceramic technology in the last decade and the extent to which ceramic materials are being increasingly considered for applications which are conventionally reserved for metals.

The engineer will detect a common thread through all the apparently diverse developments reported; ceramics are all, to some extent, brittle materials and are liable to failure by shock effects, both thermal and mechanical. The best efforts of the ceramic technologist are unlikely, in reasonable time, to permit modification of this statement other than in a matter of degree so that the engineer must, if he wishes to use these materials, design accordingly. This point of view has been accepted by nuclear design engineers and it may not be many years before ceramics are as familiar as components in marine nuclear reactors as they are now in marine boilers. The problems involved in reciprocating and rotating machinery are, of course, rather different, and here the time factor may be longer but, in the authors' view, by the joint efforts of ceramists and designers, materials which fall within the wider definition of the word "ceramic" will ultimately take their place as engineering materials.

ACKNOWLEDGEMENT

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"REED'S MARINE EQUIPMENT NEWS"

SOME MARINE APPLICATIONS OF ANTI-FRICTION BEARINGS

ACKNOWLEDMENT

This article, by J. Anderson, appeared in the March 1961 issue of "Reed's Marine Equipment News" from which it is reprinted.

BALL AND ROLLER bearings are widely used in ships for many different applications—indeed it would be difficult, if not impossible to deal with the subject adequately within the scope of one article. For this reason it is intended here to give some general information on only three of the more important marine applications of anti-friction bearings—i.e. tunnel shafts, thrust blocks and rudder stocks.

Among the obvious advantages to be gained by using anti-friction bearings are, ease of mounting, low friction coefficient and reduced maintenance together with a saving in lubricant.

Of the applications mentioned, tunnel shaft bearings are comparatively more lightly loaded. The size of the bearing selected is governed by the shaft diameter required, and this is a function of the torque to be transmitted so that bearing loads, for the most part, are related to the weight of the shaft. However, it is necessary to consider the external forces that occur and which give rise to deflections in the hull of the ship. The mal-alignment of the tunnel shaft which could result from movements of the hull would quickly cause failure of any bearing which tried to give resistance. It is therefore important that the bearings should be able to accommodate mal-alignment of the tunnel shaft.

The self-aligning feature necessary can be obtained either by using a rigid bearing, such as a cylindrical roller or taper roller bearing, in a specially adapted housing, or by using a bearing, such as the spherical roller bearing, which is self-aligning in itself and does not require a special housing.

Usually, where a rigid bearing with special housing is used for tunnel shafts, the bearing rings and cage assembly are each in two pieces and these are clamped together to give a complete bearing. It is possible with this bearing to use shafting with integrally forged flanges. Due to its more robust construction, the self-aligning bearing requires a

removable coupling and this can be of the flange or muff type.

While couplings are not really within the scope of this article, it is worth mentioning that the marine couplings associated with tunnel shaft roller bearings, although removable, are not loose. They are mounted with a fairly heavy interference (approx. .0025 in. per inch of shaft diameter) to ensure an adequate torque transmitting capacity. In spite of this heavy interference, and due to their special design, the couplings can be removed quite quickly and easily by hydraulic means. The shafts and couplings do not have to be keyed or splined so that the shaft is not weakened in any way.

Figure 1 shows a length of intermediate shafting which incorporates self-aligning roller bearings and muff couplings of the type described above. Figure 2 shows a typical bearing arrangement for tunnel shaft housings of this type. Here, a bearing with cylindrical bore is mounted directly onto the shaft although in some cases bearings with a taper bore have been mounted onto an intermediate sleeve. The housing bore limits are such that the bearing can slide easily axially in its housing so that variations in shaft length due to thermal expansion or contraction are easily accommodated.

The bearings are adequately lubricated by grease, although the grease should be chosen with care, and as it is retained in the housing a very clean installation is obtained. It is only necessary to inspect the bearings annually, at which time the grease can be replaced or replenished if necessary. It is not necessary to add grease to the housings between inspections.

It is true to say that rolling bearings are applied fairly widely to ships' tunnel shafts as can be judged by the fact that at least 250 vessels have been equipped with self-aligning roller bearings. Tunnel shaft roller bearings having a bore as large as 400 mm. (15¾ in.) are by no means uncommon

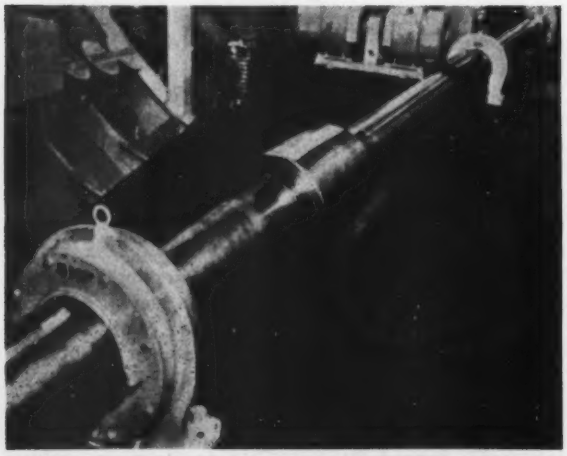

Figure 1. A section of a 300 mm. diameter shaft fitted with 10 bearings and 6 muff couplings.

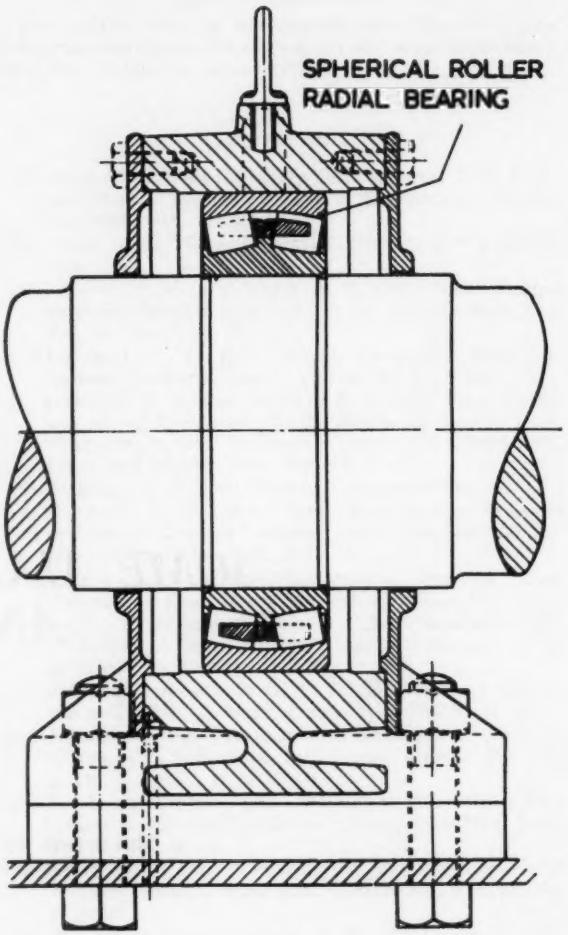

Figure 2. Cross-section of a bearing housing of the type shown in Figure 1.

and bearings as large as 500 mm. (19¾ in.) are in operation.

Anti-friction bearings also lend themselves to use in propeller thrust blocks. Those used in thrust blocks for large vessels are usually of the self-aligning thrust type, i.e. self-aligning spherical roller thrust bearing. If the major thrust load is expected to operate in both directions for approximately the same length of time, then two of these self-aligning thrust bearings are usually incorporated, but if the reverse thrust will only act for very short periods then it is usual to use one self-aligning thrust bearing in conjunction with one self-aligning radial bearing. With these bearings it is necessary to use a separable coupling and this is usually of the muff type, as previously referred to.

Figure 3 shows a bearing arrangement incorporating one spherical roller thrust bearing and one spherical roller radial bearing.

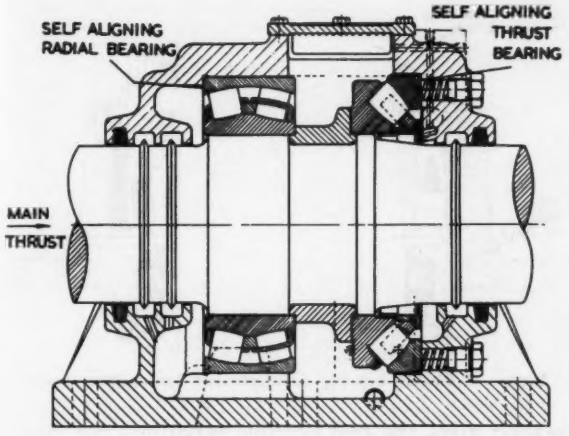

Figure 3. Larger type of thrust block used where the thrust is mainly in one direction.

While a spherical roller thrust bearing can—under certain conditions—accept radial loads, in the arrangement shown this bearing accepts axial loads only and is therefore clear in the housing bore. These thrust bearings can accept extremely high thrust loads and the arrangement illustrated is intended for applications where the thrust acts mainly in one direction. The spherical roller radial bearing accepts the reverse thrust for short periods but when this happens the springs included behind the thrust bearing outer ring ensure that this bearing is still adequately pre-loaded in order to maintain true rolling between rolling elements and rings.

These bearings are oil lubricated and special attention is paid to sealing in order to prevent oil leakage. As can be seen the felt seals are supplemented by oil flingers which are integral with the shaft. Any oil which tends to escape along the shaft is picked up and flung into the annular recesses in the housing covers from which it drains back into the housing. The distance piece between the two bearings is in two halves which are fitted into the shaft recess to be retained by the bearings when they are finally positioned. The housing itself is split on the horizontal center line to facilitate assembly.

Propeller thrust blocks of this type have been applied to shafts as large as 400 mm. (16 in.) in diameter.

For smaller and less heavily loaded applications a double-row self-aligning roller bearing is often used. The design of these bearings is such that the inclination of the roller tracks allows thrust loads to be accepted in either direction. A typical arrangement of this type is shown in Figure 4 and here the bearing is mounted direct onto the shaft and secured by means of a locknut. The bearing is grease lubricated, simple felt seals are found quite

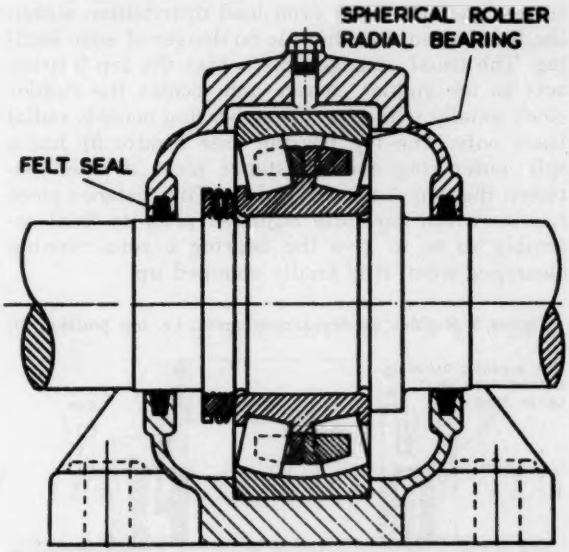

Figure 4. Thrust block of the type used on small vessels.

effective, and the housing is again split along the horizontal center line to facilitate assembly. The bearings need only be inspected annually at which time the grease should be replaced if necessary.

Quite satisfactory operation can be obtained even with the use of only one bearing as just described, for a bearing to suit a shaft diameter of 160 mm. (6¼ in.) will, for instance, accept a continuous thrust load of 4 tons to give a calculated minimum. life of 30,000 hours. Again it is necessary to use a separable coupling, but this is offset by the fact that the shaft itself is less costly as there is no need for integral flanges.

There are well over 1,000 vessels operating in all parts of the world with self-aligning roller bearings on the rudder stock. As can be appreciated, the conditions of operation which apply to rudder stocks are vastly different to those which occur with the applications mentioned previously. One of the difficulties which is encountered with plain bearings is that flattening of the bearing takes place eventually as a result of the incessant 'hammering' which occurs. This in turn results in a loose bearing. However, the extreme hardness and the toughness of anti-friction bearing components (60-62 C Rockwell) precludes the possibility of any flattening occurring. This, coupled with the high carrying capacity available, makes their use quite attractive. It is worth noting that a self-aligning roller bearing as used for a stock diameter of 10¼ in. can safely accept repeated loads of 150 tons and more.

The frequent load reversals which occur in operation are best catered for by having an assembly free from play and this is achieved by mounting the bearings so as to have a zero running clearance. Also the fact that the bearing is self-aligning en-

sures that there is an even load distribution within the bearing, so that there is no danger of edge loading. The usual arrangement is that the top bearing acts as the rudder carrier and locates the rudder stock axially while the lower bearing accepts radial loads only. The top bearing (see Figure 5) has a split outer ring and a distance piece is fitted between the two outer ring halves. This distance piece has its width carefully adjusted prior to final assembly so as to give the bearing a zero running clearance when it is finally clamped up.

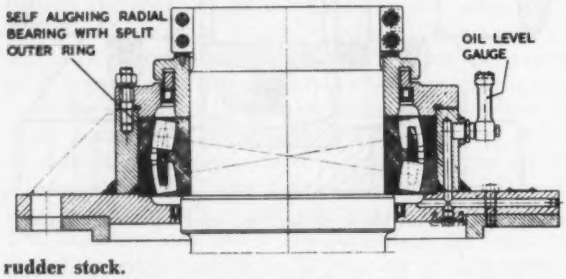

Figure 5. Rudder carrier arrangement, i.e. top position on rudder stock.

The lower bearing, see Figure 6, is mounted on a tapered sleeve and the clearance in this bearing can be eliminated by driving the bearing along its sleeve, and expanding the inner ring.

Sealing and lubrication are particularly important. Proprietary lip seals are used with both bearings on the rudder stock, but of course a more complicated bank of seals is usually applied to the lower bearing, as will be seen from the illustrations.

Both bearings are completely immersed, in oil and a pre-determined oil pressure is generated in the lower bearing housing by means of a header tank. It is usual for this oil pressure to exceed the anticipated water pressure so that any drop in efficiency of the seals will be indicated by a drop in the oil level in the header tank. The fact that the bearings are completely immersed in oil during operation gives the best possible conditions for lubrication as this always ensures an adequate supply of lubricant at the contact points between rollers and tracks. As a result of this, and the absence of clearance within the bearings, damage due to vibrations is no longer a problem.

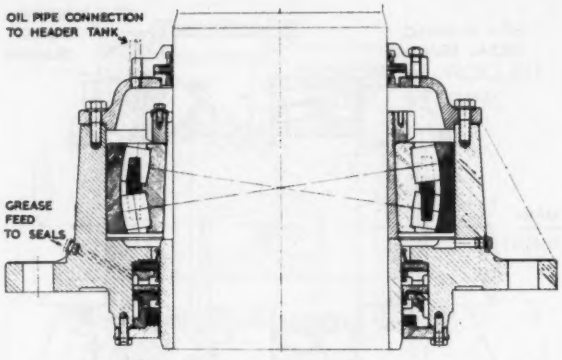

Figure 6. Lower bearing arrangement for rudder stock.

The foregoing remarks may give some idea of the extent to which roller bearings have been used for marine applications. However, it should be noted that the figures quoted concerning the number of existing applications only apply to cases where the bearings have been supplied by the organization with which the author is associated. There are, in fact, many more cases where roller bearings have been used in the applications discussed.

It should also be noted that ball and roller bearings are used extensively in pumps, motors and steering gear. In particular, self-aligning roller bearings are being used in bow-propellers and in stern-tubes, and spherical roller thrust bearings are being used in the actuating mechanism of certain types of variable pitch propeller.

In all cases the high degree of reliability associated with these bearings, together with the simple maintenance involved, is ample justification for their use.

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G. M. FAETH AND D. F. WHITE

IGNITION OF HYDRAULIC FLUIDS BY RAPID COMPRESSION

THE AUTHORS

G. M. Faeth, Research Assistant in the Department of Mechanical Engineering at The Pennsylvania State University, served as project engineer for the research program described in this paper. He received the degree of B.M.E. from Union College in 1958, and is a candidate for the M.S. degree at Penn State.

D. F. White, Associate Professor of Mechanical Engineering at Penn State, was responsible for technical supervision of the work.

A PREVIOUS PAPER published in this Journal [1] described the results of an investigation of the ignition of hydraulic fluids inside high-pressure pneumatic systems. Summarizing briefly the contents of [1], it was shown that ignition of certain hydraulic fluids could occur at a closed end of a pipe initially at low air pressure, if high-pressure air were admitted to the low-pressure section by opening rapidly a valve connecting the two parts of the system. It was also found that once ignition occurred, a flame could travel through the system, if the interior walls were coated with a film of hydraulic fluid. Several serious accidents have occurred due to this phenomenon, some of which have taken place aboard U. S. Navy vessels. The problem is therefore of particular interest to naval engineers.

The present study amplifies and extends the work reported in [1]. The principal objective of the present work was to specify ignition limits for each of several hydraulic fluids, over a range of operating conditions of a specific piping configuration. Test variables included air pressure, valve opening rate, ambient temperature, and pipe length after the valve. It was intended that the test data would serve to establish design criteria for pneumatic system components, from the point of view of eliminating the hazards due to the ignition, by rapid compression, of hydraulic fluids. Although this goal has not been realized fully, considerable progress has been made in pointing out the overall effects of some of the more important variables.

In order to introduce the work presented in the body of this paper, a brief description follows of the rapid compression process as it is presently understood. Consider a section of pipe containing a large volume of air at high pressure, connected to one side of a valve. On the other side of the valve is connected another section of the same pipe, terminating in a closed end, and initially containing air at low pressure. If the valve is opened rapidly, the pressure equalization process occurs by means of pressure waves generated during the valve action. One of the

consequences of the non-steadiness of the flow is the formation of a contact zone (a region of rapidly changing temperature and density) which separates the expanded and cooled "upstream" gas from the compressed and heated "downstream" gas. In effect, the air originally upstream from the valve acts somewhat like a cold-air piston, pushing ahead of it the air originally occupying the downstream section. At the end of the wave motion, the "downstream" gas is compressed into a small volume immediately adjacent to the closed end of the downstream pipe. Its final temperature, if heat transfer and diffusion effects are neglected would reach the adiabatic value determined by the ratio of final to initial pressures. Since, however, the wave motion occurs over a finite time interval, and since also the gradients involved are large, heat and mass transfer effects tend to lower the maximum temperature attained by the "downstream" gas. Energy transfer from the compressed gas occurs by convection and conduction to the pipe walls, and by thermal diffusion across the contact zone. Since the heat capacity of air is relatively small, a small amount of energy loss represents a relatively large decrease in temperature. It would be of considerable value to have available a theoretical analysis of the downstream gas temperature over time, as affected by the heat transfer and diffusion processes. An analysis has not been made, due principally to the total lack of information on the values of the coefficients controlling the rate processes involved. Information of this kind could probably only be obtained by direct measurement.

Aside from the fluid mechanics of the problem, the chemical and physical properties of the hydraulic fluids, particularly their auto-ignition characteristics, assume an important role. Recent data by Kennedy, et al [2] show that the self-ignition temperature of Mil-2190 hydraulic fluid (a hydrocarbon fluid) decreased from 716°F at 14.7 psia to a minimum temperature of 392°F at 1000 psia, increasing slightly to 428°F at pressures up to 4000 psia. A plot of these data is shown in Figure 1. Since the data of Figure 1 were obtained by the ordinary bomb test method, with equilibrium conditions prevailing, such data can certainly not be applied directly to the present problem. However, it is reasonable to expect that the trend in Figure 1 may appear in the test results to be described.

The hydraulic fluids used in the test program are identified in this paper by code letters A through G. Hydraulic fluid A is a hydrocarbon oil, Navy Specification Fluid 2190 TEP. The remaining fluids are synthetics, and are not named here for proprietary reasons. Inquiries concerning the chemical natures of these fluids should be directed to the sponsoring agency, with reference to Contract No. Nobs 72409.

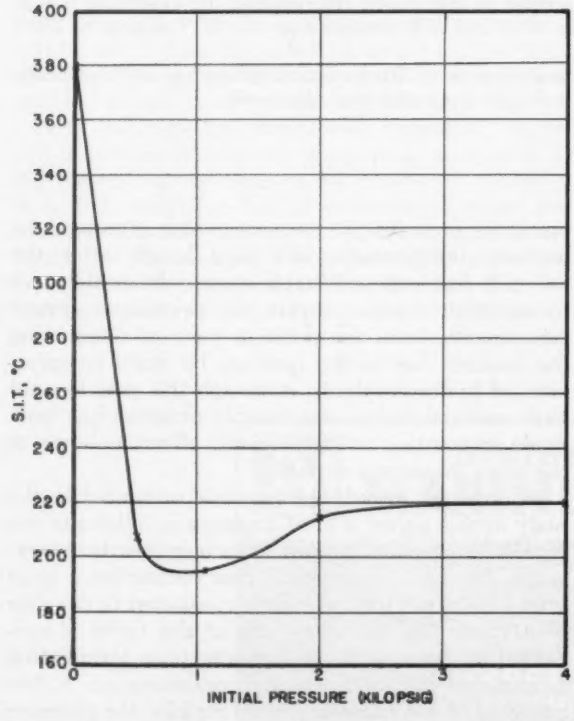

Figure 1. Spontaneous Ignition Temperature (S.I.T.) of fluid A at Elevated Pressures—Reprinted from Ref. [2].

TEST APPARATUS

A sketch of the apparatus is shown in Figure 2. High pressure air was supplied from a reciprocating compressor up to a maximum of 4500 psi. The air was filtered as it came from the compressor in order to eliminate any lubricating oil which may have become entrapped in the air during compression. The filtered air was then stored in an accumulator located on the compressor for use during testing.

The desired release pressure was obtained by use of a pressure control valve located downstream of the accumulator. Fine adjustment of the release pressure was made by closing off the supply line from

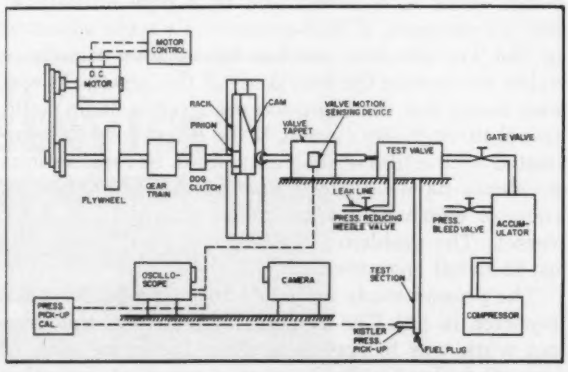

Figure 2. Test Arrangements—Room Temperature Tests.

the accumulator after the pressure upstream of the valve was slightly higher than the desired release pressure. The excess pressure was then brought to the correct value by bleeding air slowly from the high pressure section.

A sketch of the rapid opening valve is shown in Figure 3. In all cases the valve is opened by the pressure in the upstream section acting on the face of the valve plunger. For the "slow" valve opening times the motion of the valve plunger is retarded by a tappet which bears against the valve opening control cam. A valve opening control mechanism drives this cam laterally at different speeds, thereby controlling the rate of valve opening. This machine consists of an A. C. motor, which drives a counter shaft, through a set of step pulleys. The counter shaft, through a set of gears, drives the pinion, which is engaged with a rack mounted directly on the cam. An electric clutch is mounted between the pinion and countershaft. Engaging the clutch by a switch is the starting action for a test. By changing pulleys and gear ratios of this machine, numerous valve opening rates can be obtained. The range of valve opening times (from closed to fully open position) available is from 2340 to 10 milliseconds. In actual testing, however, the maximum valve opening time used was 145 milliseconds. For the "fast" valve opening rates the valve plunger was retarded only by its mass and friction between it and the valve body. Testing in that range severely limited the life of the equipment, so that it was discontinued. The minimum valve opening rate for the major portion of the test program was 10 milliseconds.

The test section was constructed of the same size tubing as the supply system, ⅝" O.D., 0.125" wall, low carbon steel tubing. The test section was closed at the end by a removable plug. The inside surface of the end plug was machined with a concave surface. The cup thus formed, combined with the vertical alignment of the test section, provided a holder for the fluid sample to be tested.

The removable end plug further served to facilitate cleaning of the test section. A similar plug placed at the top of the test section allowed a cleaning rod to be run completely through the test section. Also incorporated into the end plug assembly was an adapter for the mounting of a pressure transducer.

A system of electrical contacts was used to measure the opening rate of the valve. A contactor placed on the valve plunger was moved along an insulated strip in which was embedded two contactors, the first in a position corresponding to the start of valve opening, the second where the valve was opened fully.

When the contactors on the valve and the insulated strip were together, a D. C. circuit was closed, which deflected the beam of an oscilloscope. Successive deflections of the beam were recorded by means of a moving film camera. A timing light

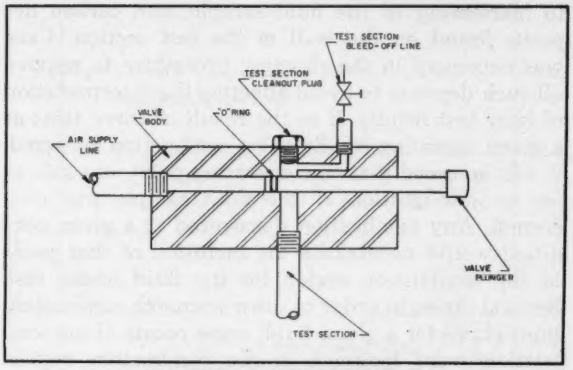

FIG.3 SKETCH OF RAPID OPENING VALVE

Figure 3. Sketch of Rapid Opening Valve.

focused on the film placed a time scale on the record. Valve opening time was then found by measuring the time between the two deflections of the oscilloscope beam.

Pressures in the test section were measured by means of a piezo-electric transducer manufactured by the Kistler Instrument Company, the pickup output being fed to an oscilloscope, and the beam trace photographed by the camera mentioned previously.

For tests at elevated downstream temperatures, the apparatus was unchanged with the exception of a constant temperature oil bath, in which the downstream section was immersed.

TEST PROCEDURES

Before the start of each testing period the end plug and the cleanout plug at the top of the test section were removed. The valve plunger was also removed and all components were thoroughly cleaned with carbon tetrachloride. A cleaning patch soaked with carbon tetrachloride was run through the valve body and test section. This operation was continued until the patch would not stain when pushed through the assembly. The valve body and test section were dried thoroughly to remove all traces of cleaning fluid.

A fresh oil sample weighing approximately 25 milligrams was then placed on the end plug, and the test section and valve were reassembled. The valve and its controlling cam were set with the gear train adjusted to give the desired valve opening rate. The release pressure was then obtained as outlined previously.

The air bleed-off from the downstream section was closed and the electric clutch was activated, opening the rapid-opening valve. After the pressure had equalized in the system, an auxiliary supply line valve located about 15 feet above the upstream end of the rapid opening valve was closed. The air was bled from the test section and the downstream end of the supply line. The end plug was then removed for inspection of the test sample.

Combustion of the test fluid was recognized due

to blackening of the fluid sample, and carbon deposits found on the wall of the test section. Care was necessary in the cleaning procedure to remove all such deposits to avoid affecting the interpretation of later test results. If as the result of three tests at a given operating condition no combustion occurred, it was assumed that this operating point was safe as far as auto-ignition of the fluid sample was concerned. Any combustion discovered at a given condition would necessitate the inclusion of that point in the combustion region for the fluid under test. Several times, in order to draw a smooth combustion limit curve for a given fluid, some points of no combustion were included in the combustion region. However, no points at which combustion did occur were included in the no-combustion region of a fluid.

Recalibration of valve opening time was made each time the valve opening rate was changed, in order to eliminate the effects of variations in the tension of the motor drive belt on the valve opening time.

At the end of testing with each length of downstream test section, measurements were made of the pressure profile at the end plug for various release pressures and valve opening times. Due to the regularity of the pressure increase at the end plug, a new variable was defined, subsequently called pressure rise rate. The pressure rise rate at the end plug was determined, as shown in Figure 4, by drawing a line approximating the mean slope of the pressure profile. The slope of this line in the units psi/msec was designated the pressure rise rate.

Pressure rise rates for the one and five foot test sections are shown in Figures 5 and 6. As would be expected, the pressure rise rate increases with decreasing valve opening time and test section length, and with increasing release pressure.

RESULTS AND DISCUSSION

A. Short Downstream Section at Room Temperature

The test results are summarized in Figure 7, where combustion limit curves are drawn in terms of valve opening time and release pressure. The area to the left of each curve represents the "no combustion"

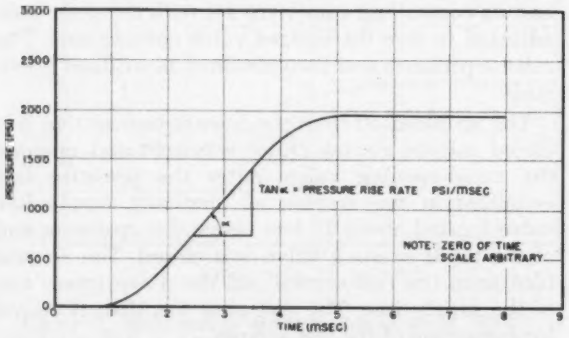

Figure 4. Typical Pressure Record at the End Plug.

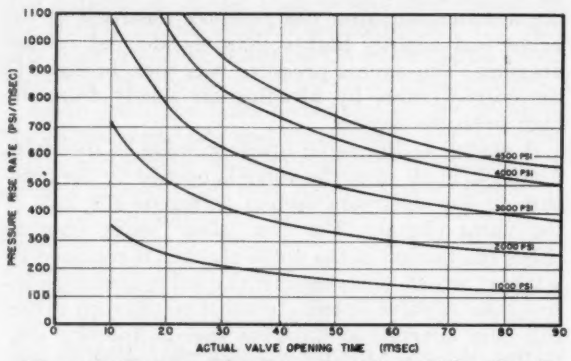

Figure 5. Pressure Rise Rate Vs. Actual Valve Opening Time 1 Foot Test Section.

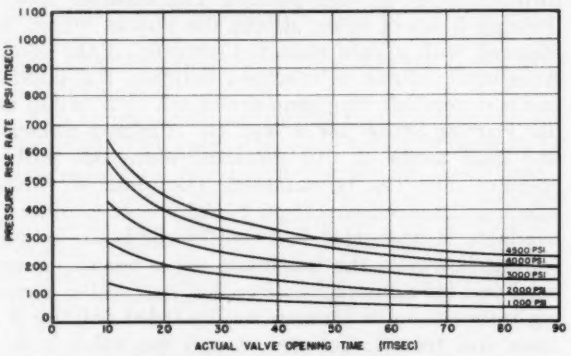

Figure 6. Pressure Rise Rate vs. Actual Valve Opening Time 5 Foot Test Section.

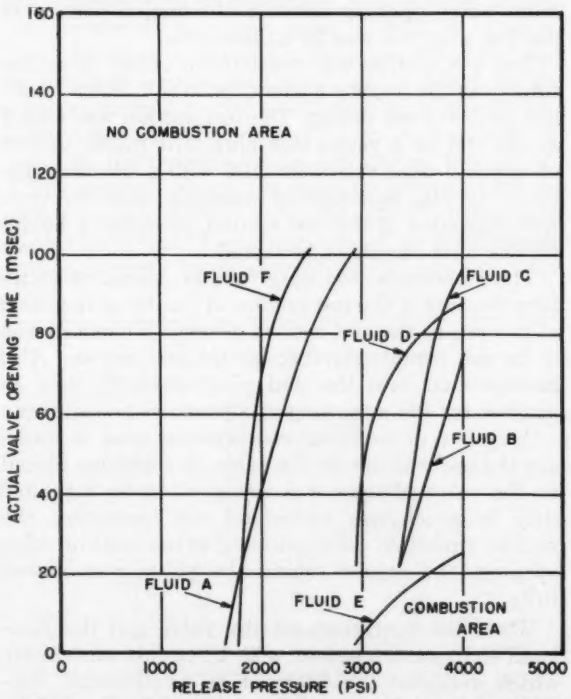

Figure 7. Combustion Limit for Fluids A,B,C,D,E,F Short Downstream Section at Room Temperature—75°F.

area for each fluid. No curve appears in Figure 7 for fluid G, since it did not ignite over the available test range at room temperature.

It is difficult to interpret the data of Figure 7, as far as any physical meaning is concerned, except to note the apparent differences in ignition resistance of the several fluids. However, as indicated by the intersections of the curves of Figure 7, it can also be seen that the relative ignition resistances of the fluids change with valve opening time. In other words, a relatively "unsafe" fluid at a fast valve opening rate may become relatively "safe" at slower opening rates. The physical reasons for such behavior are not known.

B. Long Downstream Section at Room Temperature

Fluids, A, C, and G were further tested in the five foot test section. The results of these tests are shown plotted in Figures 8 and 9. The results obtained in the one foot section with each of these fluids are also plotted for comparison purposes. It is seen that with the five foot section a higher release pressure was necessary for combustion at a given valve opening time. The curves are shown replotted in terms of release pressure and pressure rise rate in Figure 10.

From Figure 10 it is seen that the curves approach a minimum value of release pressure with increasing pressure rise rate. Some insight into this be-

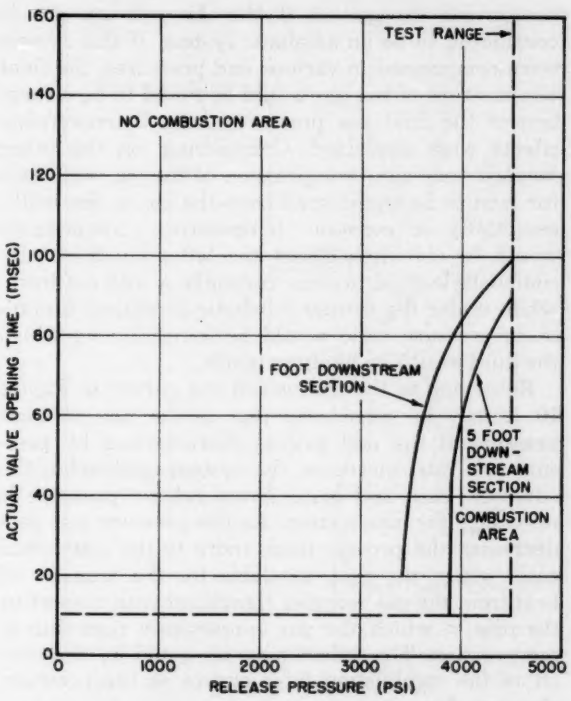

Figure 9. Combustion Limit for Fuel C with Variable Length of Downstream Section at Room Temperature—75°F.

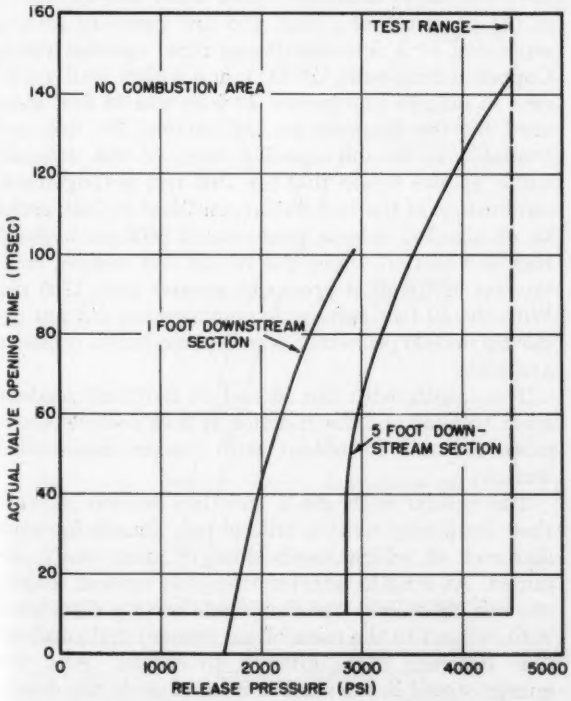

Figure 8. Combustion Limit for Fuel A Variable Length of Downstream Section at Room Temperature—75°F.

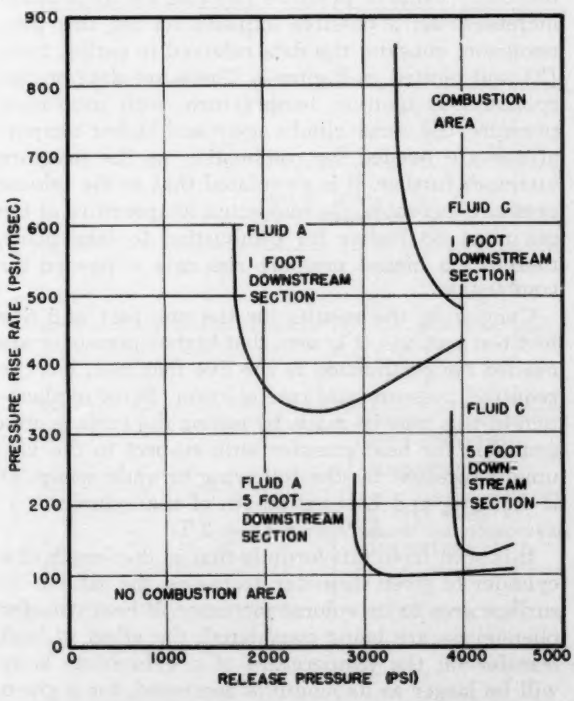

Figure 10. Combustion Limit for Fluids A & C:Vs. Release Pressure Rise Rate Various Downstream Lengths.

havior can be gained if the downstream gas is considered to be an adiabatic system. If this system were compressed to various end pressures, the final temperature of the gas would be found to be a function of the final gas pressure alone, if irreversible effects were neglected. Considering, on the other hand, a very slow compression of the gas with time for heat to be transferred from the gas to the walls, essentially a constant temperature compression would be obtained. Under the latter condition any combustible fluid present certainly would not burn, while under the former adiabatic condition, a minimum pressure ratio would be found, above which the fluid would be likely to ignite.

Returning to the analysis of the curves in Figure 10, it may be considered that as the rate of compression of the end gas, as characterized by pressure rise rate, increases, the system approaches the adiabatic case and a minimum release pressure is necessary for combustion. As the pressure rise rate decreases the process tends more to the isothermal case, where the time available for the transfer of heat from the gas becomes significant with respect to the rate at which the gas temperature rises due to compression. This effect may be noted by the cutoff of the combustion limit curves as the pressure rise rate decreases.

It is also noted in Figure 10 that as the release pressure continues to be increased above the minimum release pressure necessary for combustion, the necessary value of pressure rise rate for combustion increases. As a possible explanation for this phenomenon, consider the data referred to earlier from [2] and plotted in Figure 1. These are data on the spontaneous ignition temperature with increasing pressure, the curve climbs again and higher temperatures are needed for combustion as the pressure increases further. It is postulated that as the release pressure increases, the maximum temperature of the gas must be higher for combustion to take place, therefore a higher pressure rise rate is needed for combustion.

Comparing the results for the one foot and five foot test sections, it is seen that higher pressures are needed for combustion in the five foot case, but the required pressure rise rate is lower. Some explanation of this may be made by noting the surface area available for heat transfer with respect to the volume, expressed by the following formula where D is diameter and L is the length of the cylinder.

$$A/V = 4/D + 2/L$$

It is seen from this formula that as the length of a cylinder of given diameter decreases, the ratio of its surface area to its volume increases. If heat transfer phenomena are being considered, the effect of heat transfer on the temperature of a cylindrical body will be larger as its length is decreased, for a given time interval. Considering that the gas in the one foot test section represents a shorter cylindrical length than the five foot case throughout its entire flow history, it seems reasonable to assume that the effect of heat transfer is greater for the one foot case. Also, in the five foot test section diffusion and any mixing effects present at the contact zone are more remote from the fluid sample, thereby having less effect on the temperature of the air exposed directly to the fluid. Therefore, a higher pressure rise rate is necessary to make up for the increased heat and mass transfer effects with the shorter lengths.

Inspection of Figures 5 and 6 shows that with the longer downstream section, pressure rise rates are much lower for a given release pressure and valve opening time. Therefore, higher release pressures are necessary with the five foot test section in order to obtain the pressure rise rates necessary for combustion.

Throughout the test program it was often noted that in cases where the fluid sample did not burn it appeared to be foamy. This condition of foaminess seemed to be more prevalent in cases with the longer test section. Since, on the whole, combustion in the longer test section was more complete than in the shorter test section, weight is lent to the theory that this foaming is due to higher turbulence at the end of the tube. The higher turbulence would lend itself to more complete mixing of the fluid vapor with the air, thereby giving a more complete combustion.

It is of interest to note results given in [3] for tests similar to those described here. The apparatus of [3] consisted of a high and low pressure section separated by a solenoid driven rapid opening valve. Copper tubing with ¼″ O.D. and 0.035″ wall thickness in lengths of 5 inches, 10 feet, and 30 feet were used for the downstream test section. No data are available as to the opening time of the solenoid valve. It was found that for this test arrangement, combustion of the test fluid (identified in [3]) could be obtained at release pressures of 2600 psi in the 5 inch test section. Using the 10 foot test section, combustion occurred at pressures greater than 1200 psi. With the 30 foot test section combustion did not occur up to 3000 psi, which was the maximum pressure available.

The results with the 10 and 30 foot test sections seem to bear out the conclusion that combustion is more difficult to obtain with longer dead-ended sections.

The results with the 5 inch test section indicate that there may exist a critical pipe length for each diameter at which combustion is most easily obtained. At lengths shorter than this critical length, any mixing or heat transfer effects become significant with respect to the mass of gas present and combustion becomes more difficult to obtain. Also the energy stored in the smaller mass of air in the downstream section may not be great enough to start a combustion even though the gas temperature is high.

At lengths longer than the critical length referred to, the pressure rise rate is decreased due to the increased volume of the test section and if the pressure rise rate is sufficiently low, the auto-ignition temperature of the fluid is not reached primarily due to heat transfer effects.

C. Elevated Temperature Tests

Fluids A, C, and G were tested in this series at temperatures in the downstream section of 140 and 210 F. The combustion limit curves for these conditions are plotted in terms of release pressure and valve opening time in Figures 11, 12, and 13. The curves obtained at room temperatures for each of these fluids are also plotted for comparison.

In general it is seen that as the ambient temperature of the downstream air and the sample is increased, combustion is easier to obtain.

The results obtained with fluid G proved most interesting since it was impossible to ignite this fluid in the test apparatus at room temperature for either the 1 foot or 5 foot test sections. Upon heating the downstream section to 140 F., the combustion limit curve moves significantly to the left, greatly increasing the area in which combustion would occur. With the further increase in temperature to 210 F. the combustion limit curve shifted again, although not as drastically as for the first increase in temperature.

Fluids A and C behaved in qualitatively the same

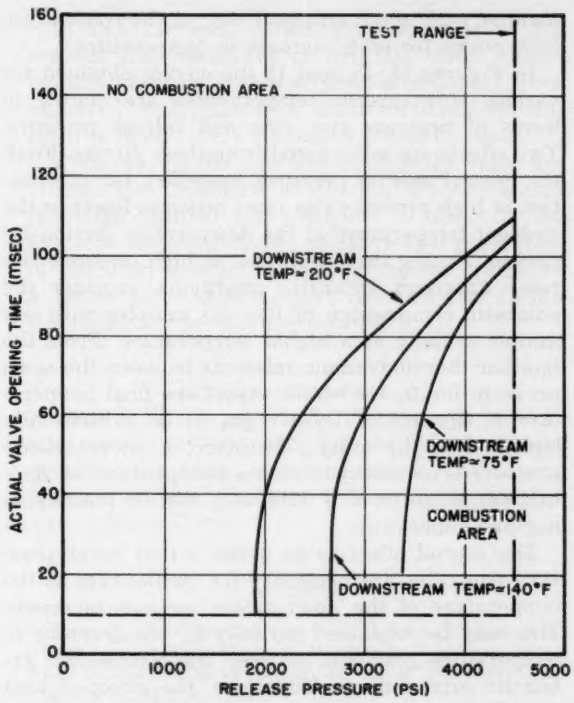

Figure 12. Combustion Limit for Fuel C at Various Downstream Temperatures.

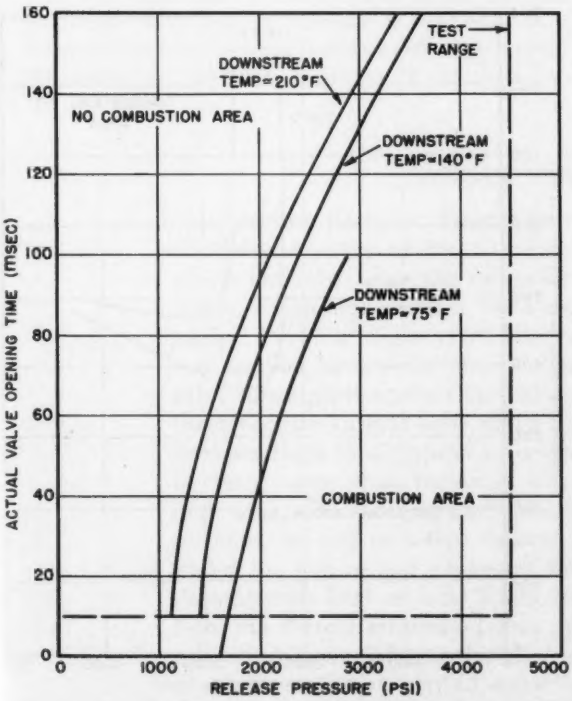

Figure 11. Combustion Limit for Fuel A at Various Downstream Temperatures.

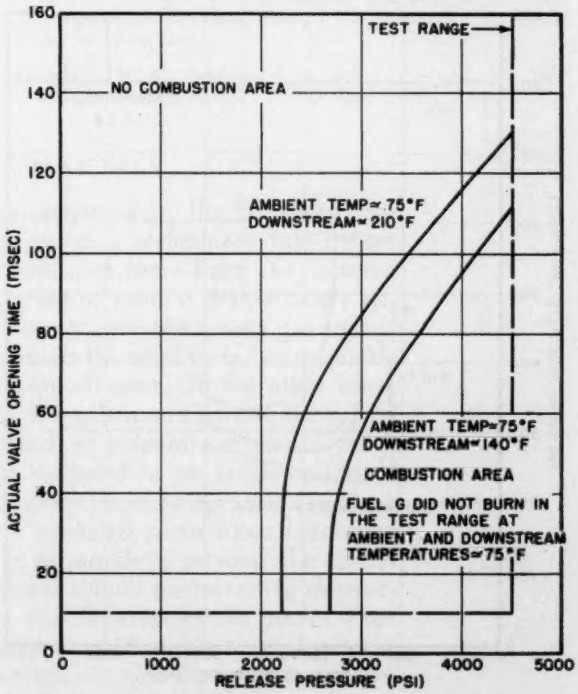

Figure 13. Combustion Limit for Fuel G at Various Downstream Temperatures.

manner with about an equal shift in the combustion limit curve for each increase in temperature.

In Figures 14, 15, and 16 the curves obtained for various downstream temperatures are shown in terms of pressure rise rate and release pressure. Two effects are to be noted from these curves. First, the critical release pressure necessary for combustion at high pressure rise rates becomes lower as the ambient temperature of the downstream section increases. Noting that conditions at high pressure rise rates approach adiabatic conditions, consider the adiabatic compression of two gas samples with one sample initially at a higher temperature. From the familiar thermodynamic relations between the same pressure limits, we would expect the final temperature of the initially hotter gas to be substantially higher than the other. Therefore, a lower release pressure is necessary to obtain the spontaneous ignition temperature of a fluid with the air initially at higher temperature.

The second effect to be noted is that lower pressure rise rates are necessary for combustion as the temperature of the downstream sections increases. This may be explained partially by the decrease in temperature gradients between the compressed gas and its surroundings. Therefore, the effect of heat transfer is reduced and the compression rate may be slower with combustion of the fluid sample still possible.

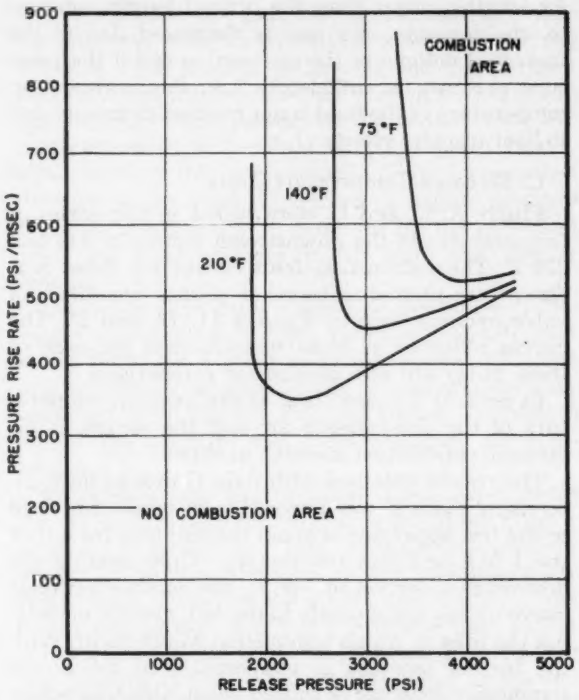

Figure 15. Combustion Limit for Fluid C: Vs. Release Pressure and Pressure Rise Rate Various Downstream Temperatures.

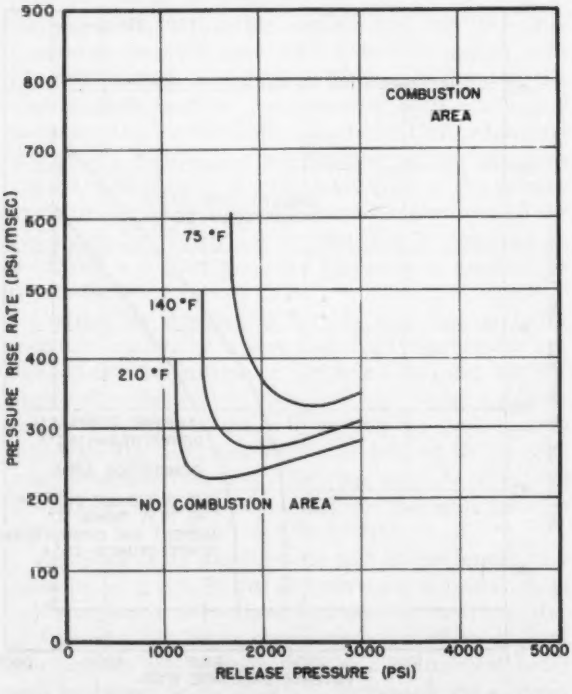

Figure 14. Combustion Limit for Fluid A:Vs. Release Pressure and Pressure Rise Rate Various Downstream Temperatures.

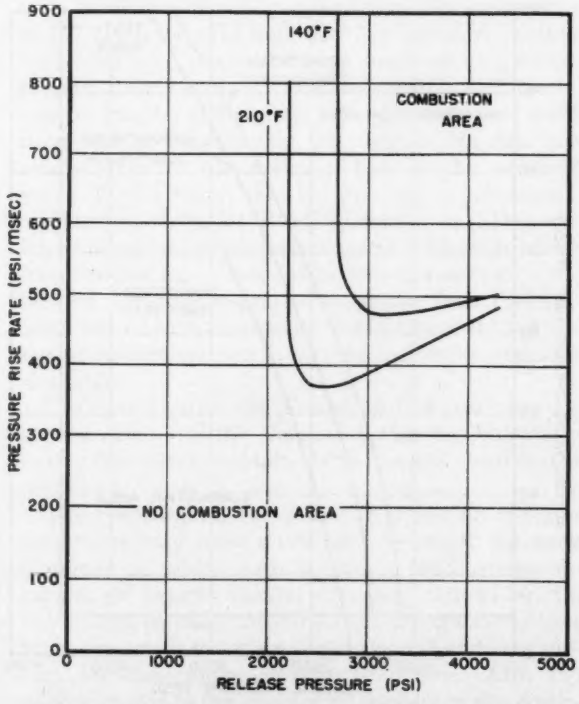

Figure 16. Combustion Limit for Fluid G: Vs. Release Pressure and Pressure Rise Rate Various Downstream Temperatures.

SUMMARY AND CONCLUSIONS

An experimental apparatus was constructed to determine ignition limits of hydraulic fluids subject to rapid compression caused by admitting high pressure air through a valve into a pipe containing air at low pressure, at the end of which the hydraulic fluid was present in the liquid phase. The compression process was controlled by varying the valve opening rate and the initial air pressure. In addition, the ambient temperature was varied up to a maximum of 210°F, and two lengths of pipe downstream from the valve were used.

The test data are presented in the form of ignition limit curves plotted on coordinates of valve opening time versus initial upstream air pressure. These curves represent the best available data on which to base the design of pneumatic system components, as far as explosion prevention is concerned.

The following conclusions have been drawn from a study of the test results:

1. For a given release pressure, the rate of pressure rise (governed principally by valve opening rate and downstream pipe length) is a critical factor influencing ignition. It is believed presently that the lower the pressure rise rate, the lower will be the maximum gas temperature due to the increased time available for energy transfer from the hot gas, and thus the less the likelihood of ignition.

2. The maximum pressure reached by the hot gas (essentially release pressure in the tests) is a critical factor influencing ignition, since it governs the maximum **possible** temperature reached by the hot gas.

3. Any increase in ambient temperature increases the likelihood of ignition, other things equal. Drastic decreases in ignition resistance were observed with relatively small increases in ambient temperature, for some of the fluids tested.

4. Increasing the pipe length after the valve decreases the likelihood of ignition, all else equal. Although ignition was observed to occur in the five foot section at lower pressure rise rates than in the short section, for a given release pressure a faster valve opening was necessary to obtain the necessary pressure rise rate.

ACKNOWLEDGEMENTS

Messrs. E. W. Beans and M. Alperstein assisted in the experimental program. Dr. A. W. Hussmann supervised the entire program and the work was supported by the U. S. Navy, Bureau of Ships under Contract Nobs 72409.

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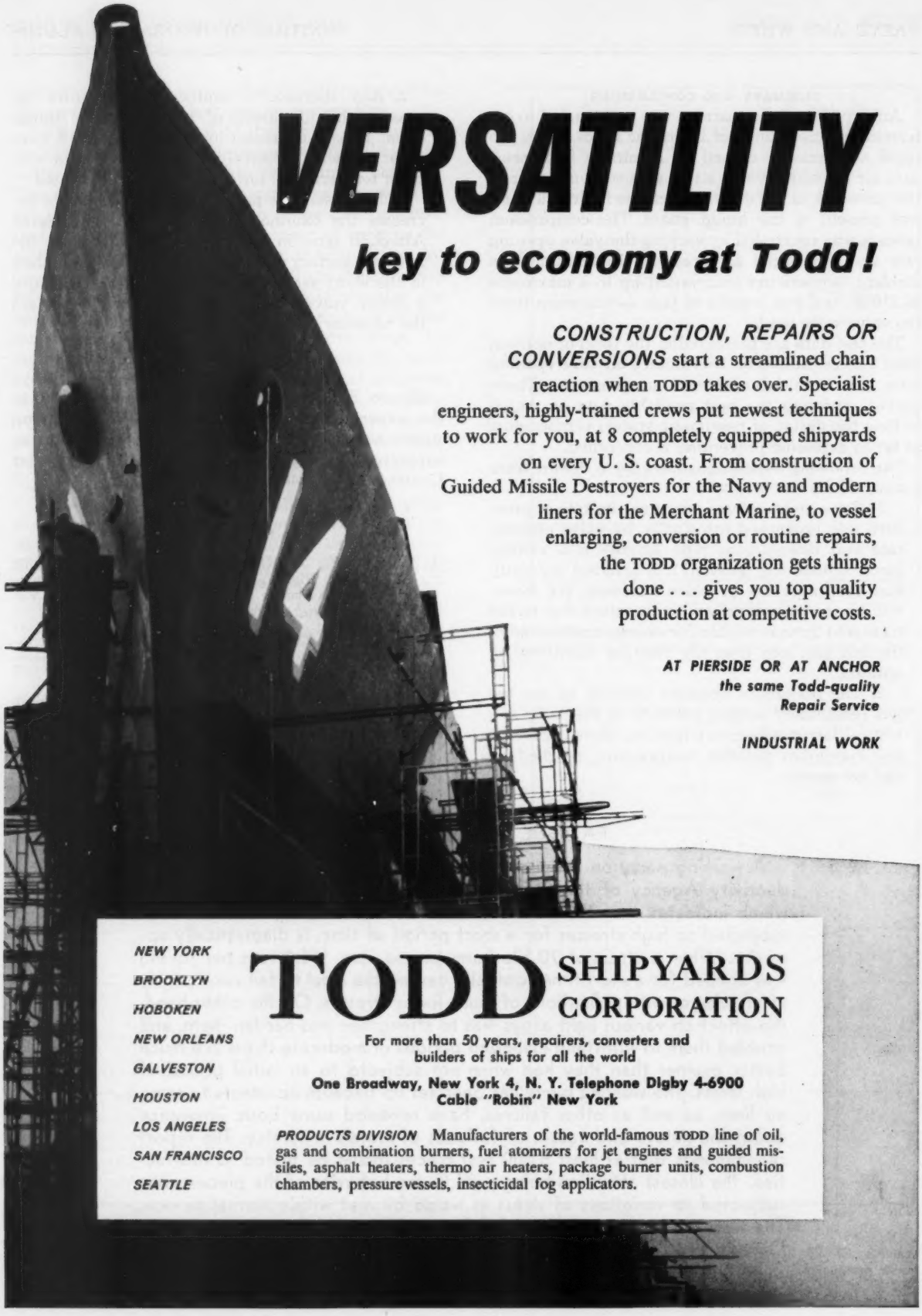
FATIGUE FAILURE IN METALS

A working party on fatigue in metals convened by the European Productivity Agency of the O.E.E.C. has issued a preliminary test report which indicates that the behavior of steel and some light alloys, when subjected to high stresses for a short period of time, is diametrically opposite. When a stress of 70 kilograms per sq. mm. (44.4 tons per sq. in.) was applied for a short time to steel it caused the steel to fail very quickly after subsequent applications of much lower stresses. On the other hand, the effect on various light alloys was to strengthen and harden them, and enabled them to withstand extended periods of moderate stress in a much better manner than they had when not subjectd to an initial period of high stress. The working party has been set up because accidents to some air lines, as well as other failures, have revealed some basic ignorance about the nature and causes of fatigues of metals in service. The report recommends that at least 5,000 test pieces should be tested to destruction, the closest attention being paid to the behavior of the pieces when subjected to variations of stress as would be met within normal service. The tests are being carried out in laboratories in nine European countries, Canada and the U.S.A. and should be completed within a year.

—SHIPBUILDING AND SHIPPING RECORD



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"THE MOTOR SHIP"

FREE PISTON ACHIEVEMENTS AND PROGRESS

ACKNOWLEDGMENT

This article, by M. E. Bartholon, chief engineer S. I. G. M. A., appeared in the March 1961 issue of "The Motor Ship" from which it is reprinted.

FOR MANY YEARS a controversial topic of discussion, the free-piston engine has been presented by its promoters as the logical result of 150 years of heat engine evolution: its flexibility is very similar to that of steam reciprocating engines, its efficiency is comparable with that of the Diesel engine, its simplicity and economy in maintenance make it a good competitor with the gas turbine, and, finally, its aptitude to continuous shaft output and operation on Bunker C fuel remind us of the steam turbine. However, such basic advantages had to be proved by actual performance in service.

As usual, for new types of prime mover, the amount of experience gained in land installations is greater than that for marine applications which concern 41 ships built so far, of which 18 are merchant vessels.

A summary of the free-piston industry seems necessary, therefore, before describing in greater detail marine achievements and future developments.

In the past three years, the total of free piston gasifiers delivered has been increased up to 500 (500,000 h.p.). Most of the recent installations are running day and night continuously throughout the year at high average load, and more than 100,000 hours of operation are registered each month, under the supervision of all classes of personnel. The accumulated total is 1,500,000 hours.

Such broad experience, coupled with close technical co-operation between S.I.G.M.A., its licensees and sub-licensees, has favored elimination of early "teething troubles" which have been reported previously [1] [2] on a number of installations.

As a first example, the life of the center ring (or combustion chamber) of the Diesel engine liner used to be as short as 2,500 hours when operating instructions were ignored; in 1960 S.I.G.M.A.-built gasifiers accumulated more than 500,000 hours' operation with only four failures of center ring—all due to errors in assembling the engine. A further example: piston wear band life is now statistically of the order of 10,000 hours when running on heavy fuel compared with 1,500 hours a few years ago, half of the saving in wear being due to the efforts of oil companies and S.I.G.M.A. in developing and selecting oils better suited to free piston service. Piston ring life of 6,000-8,000 hours is not uncommon and piston ring wear is now of the order of 0.05-0.1 mm./1,000 hr. [3]. The only other maintenance consideration of importance is the fuel injection equipment which is now simpler; the num-

TABLE I—*Free-piston-engined Merchant Ships*

Name of Vessel	Tonnage D.W.	H.P.	Service	Builder	Reversg. Arrgt.	Reduction Gear	Date Commissd.	Number of hours of service
Cantenac	1,200	1,200	Coaster	Augustin-Normand-France	Turbine	—	July, 1954	22,000
Corvo	1,200	1,200	Coaster	Augustin-Normand-France	Turbine	—	Feb. 1955	20,000
W. Patterson	10,000	6,000	Cargo ship	Bethlehem Steel General Motors	Turbine and V.P. Propeller	Milled and ground	1957	7,000
Sagitta	900	2,000	Trawler	Demag-Rickmers	V.P. Propeller	Milled	Dec. 1957	18,000
Morar	7,000	3,000	Ore Carrier	Lightgows Ltd.	Turbine	Fairfield S. B. and E. Co.	Oct. 1958	8,500
Goodwood	3,200	2,000	Cargo	Smith's Dock	Turbine	S.A.C.M.	June, 1959	6,000
Rembrandt	13,000	4,000	Cargo	Smith's Dock	V.P. Propeller	B.T.H.	Aug. 1960	200
Geestland	2,200	4,000	Banana Carrier	Werkspoor C.A.Z.	V.P. Propeller	Stoeckicht	June, 1960	2,000
Geeststar	2,200	4,000	Banana Carrier	Werkspoor N.V.S.	V.P. Propeller	Stoeckicht	Aug. 1960	500
Russian Vessels .	10,000	4,000	Cargo	U.S.S.R.	Turbine	S.A.C.M.	October, '60	350 on test
	10,000	4,000	Cargo	U.S.S.R.	Turbine	S.A.C.M.	October, '60	
	10,000	4,000	Cargo	U.S.S.R.	Turbine	S.A.C.M.	October, '60	
	10,000	4,000	Cargo	U.S.S.R.	Turbine	S.A.C.M.	October, '60	
	10,000	4,000	Cargo	U.S.S.R.	Turbine	S.A.C.M.	October, '60	
	10,000	4,000	Cargo	U.S.S.R.	Turbine	S.A.C.M.	October, '60	
	10,000	4,000	Cargo	U.S.S.R.	Turbine	S.A.C.M.	October, '60	
Robert W. Vinke	—	3,000	Whaler	A.D.M.	V.P. Propeller	Wülfel	October, '60	
Trig	—	2,000	Tug	Japan	—	—	Beginning 1961	

ber of fuel injection pumps has been reduced from three to one per cylinder, and the number of nozzles from six to three. The original pre-combustion chambers, of complex design, have been eliminated.

EXPERIENCE ON SHIPS

A peculiarity of free-piston experience in merchant ships is the fact that apart from the two original French coastal vessels, and the seven Russian vessels, all experience has been acquired on ships engined by seven licensees, contrary to what has happened in land installations. It is well known that, whatever care is taken when starting a new production, it takes time and effort to master a number of small details, not only in manufacturing but also in installation and design. Contrary to experience with land installations, only the *Cantenac*, the *Geeststar* and the Russian ships have been repeated in subsequent vessels, which means that eight different categories of vessels have been built: general cargo ships, coastal vessels, ore carriers, whalecatchers, etc. This ultimately will yield interesting results concerning the competitive position of free-piston machinery for different types of service, but it means, for the time being, more problems to solve for the installations as a whole. The same diversity is found in the solutions which have been used for expansion turbines, reversing systems and gears; and this, again, was unavoidable at this stage, in order to compare their respective merits.

As a consequence, each new ship during this early period introduced a different problem which required on a number of points new and varying solutions. It is not too surprising, therefore, that some difficulties were encountered at first on some of these new ships, which were the first installations built by the licensee and the first of that kind in the world. Two of them, however, the *Goodwood* and the *Sagitta* have achieved impressive service regularity ever since their first year of operation [5] As shown in *The Motor Ship* of February, 1959,

the *Sagitta* [4] was third in a list of 14 ships built in 1958 for the number of hours spent at sea in the first year of operation.

In the case of other ships, like the *William Patterson*, operation after a number of initial "teething troubles," has constantly improved and is now satisfactory, as a consequence of gasifier modernization. A similar example is the *Morar* and its new Mk. II turbine which will soon replace the original unit in that ship. The installation will also soon be remodelled in order to take advantage of experience gained in the first year of operation.

Rather than discuss technical details, which have been already given in previous issues of *The Motor Ship*, the tabulation (Table I) of free-piston ships in service, and the results obtained so far, gives a fair idea of the development of free-piston activities in the merchant marine. The total number of gasifier operating hours at sea is now higher than 350,000.

After reviewing the overall results in marine operation, it seems necessary to study in more detail some of the specific problems encountered in that field.

Intake Silencers.—It has been reported previously that with some earlier free-piston ships the amount of air pulsations in the engine-room was fairly high, a good part of it being due to engine-room resonance, its natural frequency being equal to engine frequency. An extensive theoretical study of the intake pulsations and of the best possible silencers has been made in France and at the same time a special experimental laboratory was set up by Alan Muntz and Co., in Great Britain, in order to facilitate model testing of all the various ways and means of eliminating such pulsations. Final tests were carried out on engines both on S.I.G.M.A. test benches in Venissieux (France) and by Smith's Dock Co., in Middlesbrough. This resulted in two solutions which have both been considered as satisfactory by the users and the visitors during recent sea trips of free-piston-engined ships. The British solution uses a fairly short intake pipe incorporating both a venturi-type of orifice and a fan. The French solution is more bulky and incorporates two pipes of about 2-meter length connecting the engine intake shrouds to a damping volume of about 3 cubic meters. This second solution is more expensive, but it gives between 7-10 per cent higher mass flow than the British arrangement, as it take advantage of the air intake pipe resonance. A marked improvement of mechanical noise level has been obtained in recent ships by using vertical sound-absorption panels.

Reduction Gears.—Two series of ships are specially interesting from the aspect of reduction gearing.

The reduction gears used on the seven cargo ships for U.S.S.R. [6] were designed by Soc. Alsacienne de Constructions Mécaniques (France). They are the first triple-reduction gears of their kind in service in the marine field. Shaft, wheels and teeth are of forged nickel chromium steel, the teeth being heat-treated and ground. The connection between the small speed shaft of the second-reduction stage and the shaft of the third-reduction stage consists of a flexible shaft. The reduction gear casing is partly welded steel and partly cast steel. The Michell thrust bearing has been designed for both astern and ahead operations and is incorporated into the reduction gear; it was dimensioned for a maximum thrust of 50 tons. The four shafts run respectively at 5,500, 1,500, 500 and 110 rpm.

The reduction gears installed in the two banana ships *Geestland* and *Geeststar* are, to our knowledge, the first of the planetary reduction gears of the Stoeckicht type now employed in merchant ships. They were built by Krupp (Germany) and have a speed reduction ratio of 17:1, the weight being about 8 tons.

REVERSING PROPELLERS

A number of recent free-piston ships have been equipped with reversing propellers. The *Rembrandt*, built by Smith's Dock, as well as the two 16,000-s.h.p. French frigates, are fitted with Kamewa-type propellers, the largest built so far in their respective countries. Blades are driven directly by an hydraulic piston located into the hub. A different design has been used on the *Geestland* and *Geeststar*. Their Werkspoor reversible propellers are mechanically driven by a long rod concentric to the shaft which is driven itself by an hydraulic piston located inside the engineroom.

SHORTCOMINGS OF SOME GAS TURBINES

The use of variable-pitch propellers has allowed, in a recent free-piston ship, elimination of astern stage losses which have been so far one of the reasons for the increase of fuel consumption. It is, however, rather difficult to ascertain with precision the resulting gain, as the corresponding turbines were of a completely different design and/or built by different turbine manufacturers. This may explain why the expected gain has not materialized with great evidence. Simultaneously, a new Mk. II turbine was developed by Rankin and Blackmore in co-operation with Power Jets, Ltd., in order to improve the Mk. I design initially installed in the *Morar*.

From preliminary tests made on the whalecatcher *Robert Vincke*, which employs the Mk. II design, it seems that an excellent solution has been found which minimizes the losses of the astern stages. A gain of 7 rpm in propeller speed has been measured when using the new masking arrangement for astern blading.

However, all the preceding remarks point out the fact that designing a marine expansion turbine, with or without astern stages, is by no means identical with designing steam or gas turbines, as operating temperatures, pressures and densities are very different. The lack of proper turbine development can

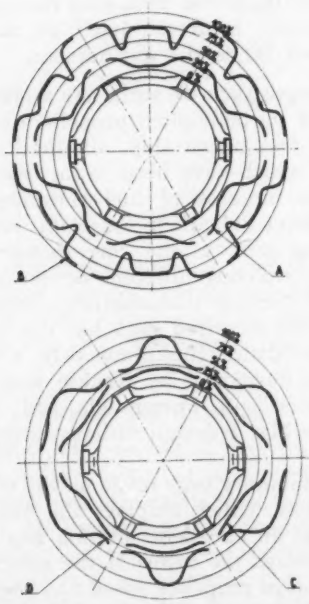
certainly be blamed for some of the relatively poor consumption obtained in early free-piston ships. It has generally been found that it is only after two or three units that turbine builders have been able to achieve the initial guarantee figures: it is quite logical to think that the best possible results have not yet been achieved and that further gains can be expected with future developments. The same remarks apply to prices; in the total cost of a free-piston propulsion unit more than half is devoted to the price of the expansion turbine and gears; and this, by itself, is enough to explain why first costs of free-piston machinery are of the same order as slow-speed Diesel costs in spite of the fact that their weight is between two and four times less. It is expected that, as the number of free-piston ships increases, turbine builders will be able to rationalize their production and to produce expansion turbines, more specifically designed for free-piston installations and take full advantage of the very low admission pressure and temperature.

FUTURE DEVELOPMENTS

Until two years ago, the major objective of free-piston research was towards achieving full reliability of the original design; since that time basic research has been directed towards increasing the rating of the engine without increasing mechanical and thermal stresses of parts. And this has now been successfully achieved.

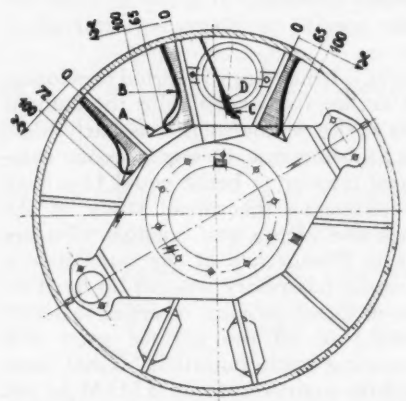
On every basic part, experimental stress studies have allowed gains of maximum stresses of from 35 to 55 per cent (Figs. 1 and 2) which concern the center section (combustion chamber) and compressor piston. Studies on combustion have resulted in better flame distribution (Figure 3) and lower exhaust temperatures; a new and cooler piston obtained by detail modifications of the earlier model has been in production for two years. As a result, smokeless and continuous operations at 20 per cent overload for more than 1,000 hours have been recently achieved.

The absence of oblique thrust on the piston and the independence from crankshaft connection allows a higher piston speed than with Diesel engines. Higher piston speeds are under active experimental consideration.



A—C before clamping. B—D after clamping. It will be noticed that in the improved design the stresses at the critical point (prechamber seat) have been decreased by 25%. In present design where prechambers have been eliminated the gain is larger. The actual gain in service is still larger because maximum mechanical stresses now appear where heat stresses are minimum.

Figure 1. Stresses of center section with the original design (top) and present design (below).



A—Stress in the most critical rib (corresponding to C design). B—Actual stresses corresponding to D design. For the sake of presentation C and D have been shown at 90° from original position.

Figure 2. Compressor piston—stresses before and after experimental stress study.

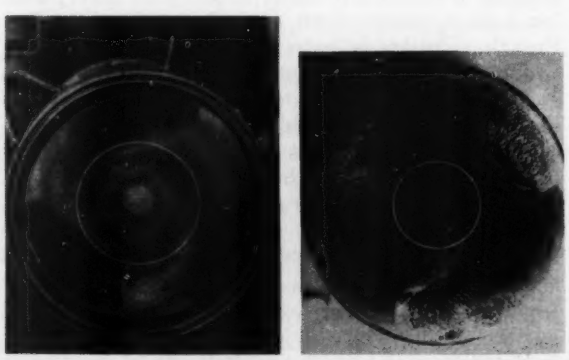


Figure 3. Engine piston with original nozzle (left) and with actual nozzle (right). A special process has been used to materialize the flame contour on piston head. It will be seen that the core of air in the center unaffected by combustion is now much smaller.

In parallel, systematical studies of the inlet arrangements in conjunction with S.I.G.M.A.'s main British licensee Alan Muntz and Co., have permitted a large increase of compressor air flow (15 per cent) and a corresponding increase in output. Preliminary tests on intercooling between compressor and Diesel cylinder have shown the feasibility of the scheme for free-piston engines. The same gains which have been already obtained in recent years from Diesel engines thanks to intercooling are now possible and will be soon obtained in free-piston engines.

THE NEW GS.2.34 GASIFIER

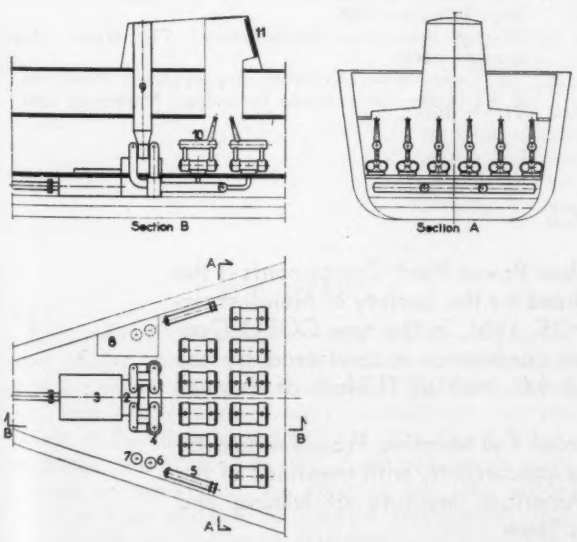
The official announcement of the new GS.2.34 model comes only after more than 4,300 hours of test have been carried out on four prototypes in research laboratories of General Motors Corps. in Chicago and S.I.G.M.A. laboratories in Venissieux.

This new machine has been obtained by grouping together two GS.34 gasifiers. Main characteristics are: all the parts are identical to similar parts of the GS.34; moving parts are operating 180° out of phase and remain so within ±2° independent of load or load changes. In such a way, the compressor piston of the gasifier A delivers scavenging air at a time when scavenging ports of gasifier B are opened, which results in a net reduction of delivery losses due to storage in engine case.

Engine case temperatures are, therefore, 17° C. lower, which makes possible an increase of 10-15 per cent in nominal rating compared with the single-cylinder rating. Fuel consumption is about 5 per cent less. Furthermore, it is very easy to fit an intercooler in the passages between the two scavenging cases and this will later result in further gains in output without increase of thermal stresses. Additionally, the fact that inlet and exhaust pulsations are dephased by 180° makes possible a very large reductiton of dampening intake and exhaust volumes down to one-fourth of the original size.

Finally, the reduction of occupied volume, due to the use of the twin-cylinder design, is 40-50 per cent, and the reduction in weight of the order of 25 per cent.

This, together with the already described new schemes for reversing permits very attractive propulsion units to be designed for up to 130,000 s.h.p. per shaft (Figure 4).



1—Gas generator. 2—Turbine. 3—Reduction gear. 4—Reversing valves. 5—Coolers. 6—Water pumps. 7—Lub. oil pumps. 8—Control position. 9—Turbine exhaust. 10—Air inlet manifold. 11—Air filters.

Figure 4. Diagrammatic arrangement of a 22,000-s.h.p. installation for a tanker.

EFFICIENCY

One of the most widely circulated errors about the free-piston engine is the belief that its efficiency will never be as good as that of the Diesel engine. This would not be a decisive handicap, as a lower efficiency has never stopped sales of steam turbines, but thermodynamics, however, emphasize that theoretical efficiency of a cycle is basically fixed by maximum cyclic temperature and overall compression ratio, which are as high, if not higher, in free-piston engines than corresponding values for Diesel engines.

Explanation for the present difference in consumption derives, therefore, from temporary practical considerations. First, in marine service, gears and reversing stages are known to have given losses as high as 12 per cent in early designs, which in the second generation of turbo-reducers were reduced to 6-7 per cent and should eventually represent no more than 3-4 per cent with well masked reversing blades and Stoeckicht-type gearing.

A second factor is turbine efficiency; contrary to some existing opinions, expansion in a turbine is basically as efficient as expansion in a cylinder with crankshaft assembly, and figures as high as 91 per cent have been obtained for aircraft gas turbines while 89 per cent has been measured on free piston expansion turbines. But such figures have never been obtained, with but one exception, before Mk. IV models, and represent the outcome of a fair amount of development.

But the most basic reason for today's moderate free-piston efficiency is the fact that the present gasifier uses five times as much air as is needed for stoechiometric combustion and, therefore, transfer aerodynamic losses are five times higher than theoretically necessary, and the mechanical efficiency is lower than it should be.

The reasons for this are easy to understand from Figure 5. Work W_C necessary for compression of one pound of air versus compression ratio is given by curve C. Work obtainable by stoechiometric

combustion of one pound of air in the engine cylinder is W_E given in curve E. To operate according to free-piston principle, W_C must be equal to W_E, but this is only possible at a high supercharging ratio of the order of 8 and today's technology does not provide ways and means to run at such a value; 3.8 was initially found as a possible maximum.

It was, therefore, necessary both to increase the work of compressor by compressing R=2.5 times one pound of air instead of one in the compressor (curve C'); (0.5) in engine cylinder (curve E") and to decrease W_E by using a lower fuel air ratio and as seen before the product:

$$2.5 \times \frac{1}{0.5} = 5.$$

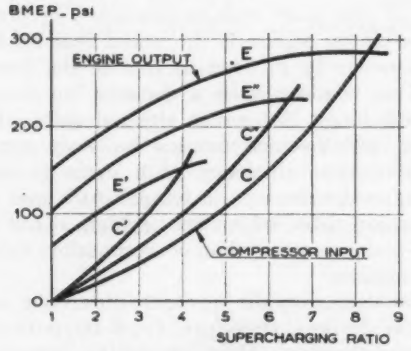


Figure 5. Energy balance in a free-piston engine.

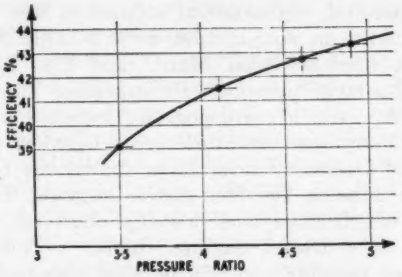


Figure 6. Efficiency versus supercharging pressure ratio. The rapid increase of efficiency versus pressure ratio will be noted.

But, as experience is gained, it becomes possible to run at a progressively higher supercharging ratio: 4.2 in today's production engines and 4.8 in 1960 for experimental S.I.G.M.A. free piston engines. This trend is now being accelerated by taking advantage of intercooling. It will be appreciated that with continued research and development there must follow appreciable gains in efficiency as the specific pumping and mechanical losses are consequently decreased.

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NOTICE

Inspection of Steam Power and Nuclear Power Plant Components is the subject of a one-day conference scheduled by the Society of Nondestructive Testing for Wednesday, October 25, 1961, in the new COBO Convention Hall in Detroit, Michigan. The conference is sponsored by the Society's Utilities Committee on which Mr. Helmut Thielsch of Grinnell Company serves as Chairman.

The conference is a part of the Annual Fall Meeting Program held by the Society for Nondestructive Testing concurrently with meetings of the American Society for Metals, the American Institute of Mining and Metallurgical Engineers and the Metals Show.

B. V. KORVIN-KROUKOVSKY

*DIRECTIONAL STABILITY AND STEERING OF SHIPS IN OBLIQUE WAVES**

THE AUTHOR

received a Sc.M. degree in Aeronautical Engineering from the Massachusetts Institute of Technology and for many years was connected with aircraft industry as Vice President in Charge of Engineering at EDO Corporation. Subsequently, he spent several years as the Research Professor in Fluid Mechanics at Davidson Laboratory, Stevens Institute of Technology, where he was drawn into the problems in the theory of Naval Architecture. He is now retired but maintains consulting connection with Davidson Laboratory. He is the author of many articles and several papers delivered at the Society of Naval Architects and Marine Engineers on ship motions in waves and on propulsion characteristics of ships, and is active in Technical Panels of this Society.

INTRODUCTION

THE MOST COMPLETE analysis, available to date, of the rudder and ship-yawing motions in oblique irregular waves was made by Rydill [1]. It was based on, and represented further development of, the work of Davidson and Schiff [2], Schiff and Gimprich [3], and Weinblum [4]. These references were developed specifically for ships at sea. The analysis of a body stability in initially undisturbed fluid was developed earlier by Bairstow et al [5] in connection with airships.

The objective of this paper is to present a unified picture of the directional stability and steering of ships in oblique waves. The basic concepts are abstracted from the foregoing references, so that needless to say, the study of these references is necessary for the complete understanding of the subject. It is however, a difficult study because different authors stress different aspects of the problem. Also, some use mathematical techniques not familiar to most naval architects, and there is not yet a universal system of notation. These complications often make it difficult to concentrate on the continuity of the underlying thought, and consequently make it desirable to present a brief exposition of the basic principles, devoid of all but immediately relevant details. This exposition will be presented under four headings:

* Lecture delivered on June 15, 1960, as part of the Davidson Laboratory Seminar on the Behavior of Ships at Sea, Stevens Institute of Technology, Hoboken, N. J.

Directional stability of a ship in smooth water.
Forces and moments caused by regular long-crested waves.
The effect of rudder action.
Motions of a controlled ship in waves.

Notation similar to that recommended in [6] will be used.

DIRECTIONAL STABILITY OF A SHIP IN SMOOTH WATER

Consider a ship, traveling in smooth water on a certain prescribed heading; as a result of a certain initial disturbance, the heading is changed instantaneously by an angle θ, and a certain lateral velocity v is imparted to the ship (Figure 1). The vectoral addition of the lateral velocity v and the forward velocity V produces an angle of yaw between the resultant velocity and the ship's plane of symmetry. This angle of yaw is accompanied by a lateral hydrodynamic force Y and hydrodynamic yawing moment N. It is now desired to evaluate the ship motions which result from these disturbances.

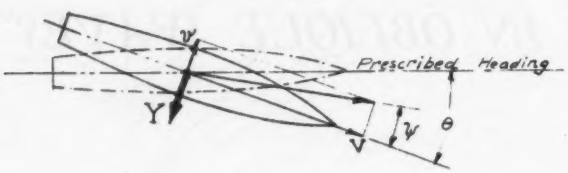

Figure 1.

The mathematical analysis invariably starts with the statement of that law of Newton which equates the force acting on a free body to the change in momentum of the body. For the combined lateral and yawing motion this is expressed by two equations

$$\begin{aligned} M(\dot{v}+V\dot{\theta}) &= Y \\ I_z\ddot{\theta} &= N \end{aligned} \qquad \ldots\ldots\ldots\ldots (1)$$

where M is the ship's mass

I_z is the ship's moment of inertia about the vertical axis and dots designate differentiation with time, i.e., $\dot{v}=dv/dt$, $\ddot{\theta}=d^2\theta/dt^2$. The first term, $M\dot{v}$, on the left hand side is self-evident, and the second term, $MV\dot{\theta}$, results from the change of direction of the forward moment vector.

A rigorous mathematical analysis of hydrodynamic forces is possible in the case of ellipsoids moving in an unbounded frictionless fluid, in which case both the force Y and the moment N are caused entirely by the acceleration of the fluid. Lamb has shown that in this case the force and moment can be expressed in terms of the coefficient of accession to inertia. By assuming the behavior of a ship on the water surface to be similar to that of submerged ellipsoids, equation (1) can be rewritten as

$$\begin{aligned} M(1+k_y)\dot{v}+M(1+k_x)V\dot{\theta} &= Y' \\ I_z(1+k_{zz})\ddot{\theta} &= N' \end{aligned} \qquad \ldots\ldots\ldots\ldots (2)$$

where k_y is the coefficient of accession to inertia in the lateral direction, k_x that in the longitudinal direction and k_{zz} is the coefficient of accession to the moment of inertia about the vertical axis. These coefficients represent parts of the total hydrodynamic force and moment which are caused by fluid inertia. The remaining force Y′ and the moment N′ are caused by the viscosity of the water, and depend on the angle of the yaw Ψ. In fluids of small viscosity, such as air and water, the effect of viscosity is limited to a small region, known as the boundary layer. Outside of this region the fluid flow can be treated as potential. The action of the viscosity in the present case is to trigger the generation of the yaw-dependent force $Y'(\Psi)$ and moment $N'(\Psi)$ in the same sense as the viscosity causes the occurrence of circulation (a potential motion) in airfoils and the resultant generation of a lift force.

Equations (2) appear to contain three variables v, θ, and Ψ, but these are not independent and can be reduced to two variables by observing that

$$\begin{aligned} v &= -V\sin\Psi \cong -V\Psi \\ \dot{v} &= -V\dot{\Psi}\cos\Psi \cong -V\dot{\Psi} \end{aligned} \qquad \ldots\ldots\ldots\ldots (3)$$

assuming the disturbance angles Ψ and θ to be small.

Equations (2) are put in non-dimensional form by dividing the force equations by $\frac{1}{2}\rho AV^2$ and the moment equation by $\frac{1}{2}\rho ALV^2$, where A is the projected lateral ship area and L is the ship's length. Furthermore, the differentiation with respect to time is replaced by differentiation with respect to the path traveled. This is expressed in term of the ship's length, so that, for instance,

$$\dot{\theta}=\frac{d\theta}{dt}=\frac{d\theta}{ds}\frac{ds}{dt}=(V/L)\theta' \qquad \ldots\ldots\ldots\ldots (4)$$

where prime indicates the differentiation with respect to the path s.

Making the above substitution in equations (2) leads to

$$\begin{aligned} -m_2\Psi'+m_1\theta' &= Y'/(\tfrac{1}{2}\rho AV^2) \\ I\theta'' &= N'/(\tfrac{1}{2}\rho ALV^2) \end{aligned} \qquad \ldots\ldots\ldots\ldots (5)$$

where

$$m_1=\frac{M(1+k_x)}{\frac{1}{2}\rho AL}$$

$$m_2=\frac{M(1+k_y)}{\frac{1}{2}\rho AL}$$

$$I=\frac{I_z(1+k_{zz})}{\frac{1}{2}\rho AL^3}$$

The non-dimensional coefficients of hydrodynamic force of viscous origin on the right hand sides of equations (5) can be expanded into sums of partial derivatives

$$\begin{aligned} Y'/(\tfrac{1}{2}\rho AV^2) &= \Psi Y_\psi+\theta' Y_{\theta'} \\ N'/(\tfrac{1}{2}\rho ALV^2) &= \Psi N_\psi-\theta' N_{\theta'} \end{aligned} \qquad \ldots\ldots\ldots\ldots (6)$$

where Y_ψ, for instance, $=\partial Y/\partial x$ is the slope of the

non-dimensional transverse force plotted against yaw angle Ψ. The primes of the coefficients on the right hand side are omitted, as it will be generally understood that they are used in non-dimensional form indicated by the left hand side of Equation (6).

Substituting equations (6) into (5) and transposing, the following is obtained

$$\begin{aligned} m_2\psi'+\psi Y_\psi-\theta(m_1-Y'_{\theta'})&=0 \\ I\theta''-\Psi N_\psi+\theta' N_{\theta'}&=0 \end{aligned} \quad \ldots\ldots\ldots\ldots\ldots\ldots (7)$$

These are the final differential equations for a ship in smooth water with rudder fixed.

The solution of equations (7) has the form

$$\begin{aligned} \Psi&=\Psi_1 e^{p_1 s}+\Psi_2 e^{p_2 s} \\ \theta'&=\theta'_1 e^{p_1 s}+\theta'_2 e^{p_2 s} \end{aligned} \quad \ldots\ldots\ldots\ldots\ldots\ldots (8)$$

where $\Psi_{1,2}$ and $\theta'_{1,2}$ represent the magnitudes of the initial disturbance, s is the distance along the ship's path measured in ship lengths, and $p_{1,2}$ are the coefficients indicating the rate of growth or decay of the initial disturbance. One of these, let us say, p_2, has usually a large negative value, indicating a rapid decay of the disturbance motion, and is therefore of no importance. The sign and the magnitude of the other, p_1, determines whether the ship is stable or unstable. A positive value of p_1 indicates a growth of any initiated disturbance of the original ship's path and heading, i.e. the directional instability. A negative value of p_1 indicates that after each disturbance, the angle Ψ and the angular velocity θ' decay and eventually vanish. In this case the ship returns to a straight path without any action from the helmsman. Attention should be called to the fact that equations (8) define Ψ and θ', *but not the heading angle* θ. Vanishing Ψ and θ' indicate the return to a straight path but not to the original heading θ, i.e. the course of a stable ship may (and to some extent does) change as a result of a disturbance.

Substitution of equations (8) into (7) gives the solution for the coefficients p in the form

$$p_{1,2}=-a\pm\sqrt{a^2-b} \quad \ldots\ldots\ldots\ldots\ldots\ldots (9)$$

where

$$a=\frac{IY_\psi+m_2N_{\theta'}}{2m_2I}$$

and

$$b=\frac{Y_\psi N_{\theta'}-m_1N_\psi+N_\psi Y_{\theta'}}{m_2I}$$

So far various stability derivatives in the foregoing equations were obtained experimentally by testing models on a rotating arm or with an oscillator. Theoretical methods of computing these coefficients on the basis of a ship's form as yet have not been developed. Neither the scale effect on hydrodynamic forces, nor the free surface effects on hydrodynamic masses have been considered.

FORCES AND MOMENTS CAUSED BY REGULAR LONG-CRESTED WAVES

Calculations of the lateral and yawing moment caused by waves usually have been made on the basis of the Froude-Kriloff hypothesis. In this hypothesis it is assumed that hydrostatic pressures in water are modified by the orbital water motion in waves but are not affected by the flow disturbance caused by the presence of a ship. Waves are assumed to be long in comparison with the beam of a ship. This last assumption eliminates the difficulty of pressure integration along the periphery of a ship's section, and permits the evaluation of the lateral force as the product of the ship sectional area A and the horizontal pressure gradient in water, $\partial p/\partial y$. The water pressure in waves is given by the expression

$$p=a\rho g e^{kz}\cos[k(x'-ct)]$$

where a is the wave amplitude, $k=2\pi/\lambda=\omega^2/g$ and x' denotes the distance in the direction of wave propagation. The origin $x'=t=0$ is taken at an instantaneous position of one of the wave crests. The vertical coordinate z is taken as positive upwards, i.e. it has a negative value over the submerged volume of a ship. This expression is converted to the ship coordinates (on the basis of Figure 2) by letting

$$x'=x\cos\chi+y\sin\chi$$

giving

$$p=a\rho g e^{kz}\cos[k(x\cos\chi+y\sin\chi-ct)]$$

The transverse component of the pressure gradient at the centerline of a ship (y=0) becomes in ship coordinates

$$\partial p/\partial y=-a\rho g k e^{kz}\sin\chi\sin[k(x\cos\chi-ct)] \quad \ldots\ldots (10)$$

Ship speed does not enter into the foregoing expression since the pressure is (erroneously) assumed to depend only on wave properties, and not to be affected by the action of the ship on waves.

The lateral force exerted by waves on a ship is

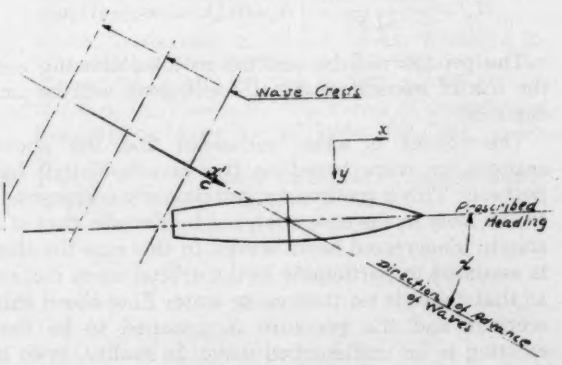

Figure 2.

$$Y_w = \int_L A_x (\partial p/\partial y)\,dx$$

$$= -a\rho g k e^{-kz_o} \sin\chi \int_L A_x \sin[k(x\cos\chi - ct)]\,dx \quad \ldots\ldots (11)$$

and the yawing moment is

$$N_w = -a\rho g k e^{-kz_o} \sin\chi \int_L A_x x \sin[k(x\cos\chi - ct)]\,dx \quad \ldots (12)$$

in writing the above expression the absolute value of the depth z_o of the effective wave streamline is taken to be at the center of buoyancy, corresponding to the effective wave slope as defined by Froude. It will be noted that the product ak in the above expression is the maximum wave slope, $(\partial\eta/\partial x')_{max}$.

In harmonic waves the variation of the force Y_w and the moment N_w also will be harmonic and it is only necessary to calculate the amplitude of their fluctuations. For the moment N_w, assuming a ship to be symmetrical fore-and-aft, this is accomplished by letting t=0 in equation (12). This places the wave crest intersection with the plane of symmetry at the midship station. The maximum positive and negative wave slopes will, therefore, be located at forward and after ship sections, thus causing the maximum yawing moment, with zero lateral force.

The largest lateral force, Y_w, will occur when the maximum wave slope i.e. wave nodal point, is at the midship section. Mathematically this is represented in equation (11) by changing sin (kx cosχ) to cos (kx cosχ) and by letting t=0. At this instant the yawing moment N_w is zero in regular long crested waves. The wave-caused force and moment must be non-dimensionalized for use with the equations of motion (7). This is accomplished by dividing equation (11) by $\frac{1}{2}\rho AV^2$ and equation (12) by $\frac{1}{2}\rho A\,LV^2$. Introducing also the Froude number $F^2 = V^2/gL$, the following non-dimensional expressions are obtained:

$$Y_w' = \frac{2\,ake^{-kz_o}\sin\chi}{AF^2} \int_L A_x \sin[k(x\cos\chi - ct)]\,dx$$

$$\ldots\ldots (13)$$

$$N_w' = \frac{2\,ake^{-kz_o}\sin\chi}{ALF^2} \int_L A_x x \sin[k(x\cos\chi - ct)]\,dx$$

The primes will be omitted in the following and the use of non-dimensional coefficients will be understood.

The reader is again reminded that the above expressions were based on the Froude-Kriloff hypothesis. This hypothesis approximately corresponds to the facts in the case analyzed by Froude, that of a ship in longcrested beam waves. In this case the ship is assumed to participate in the orbital wave motion so that there is no transverse water flow about ship sections and the pressure is assumed to be that existing in an undisturbed wave. In reality, even in this case there remains the disturbed flow in the vertical direction which affects the pressure distribution. In oblique waves, different ship sections find themselves in different parts of the wave profile, the existence of the transverse water flow is unavoidable, and should not be neglected. No complete analysis of this situation has been made, but on the basis of the work of G. I. Taylor [7] and Korvin-Kroukovsky and Jacobs [8] it can be expected that expressions (11), (12) and (13) will be modified by inserting the factor $(1+k_y)$ next to A_x under the integral signs. The evaluation of the ship speed effect on the hydrodynamic force and moment will require, however, the development of a more complete analysis. Such an analysis was made, for submerged spheroids only, by Havelock [9] and Wigley [10].

THE EFFECT OF RUDDER ACTION

It was already mentioned that "directional stability" causes a ship to return to a straight path after each accidental disturbance, but generally on a new heading. The use of the rudder is needed to restore the specified heading, and in particular this becomes necessary with repeated disturbances caused by waves. A displaced rudder causes the lateral force and yawing moment designated by Y_δ and N_δ. At the present time these can be evaluated only experimentally by model tests. It will be understood that the lateral force and yawing moments will be non-dimensionalized by dividing the measured quantities by $\frac{1}{2}\rho AV^2$ and $\frac{1}{2}\rho ALV^2$ respectively, and the primes, usually indicating this action, will be omitted.

It should be emphasized that Y_δ and N_δ are generally larger than can be estimated from the force acting on the rudder itself, since the displacement of a rudder affects the pressure distribution over the adjacent ship areas. In particular, in the case of a rudder hinged at the end of deadwood or a large skeg, the total force Y_δ is about double that acting on the rudder itself.

The motions of a rudder-controlled ship are defined by inserting Y_δ and N_δ in place of zero on the right hand sides of equations (7). It is necessary, however, to add a third equation defining the rudder angle δ as a function of the change of heading θ. The actions of a human helmsman are too complicated, and it is necessary to assume or specify a certain type of automatic control. The most elementary of these is the one in which rudder angle δ is proportional to the deviation θ from the prescribed heading, i.e. $\delta = \alpha\theta$, where α is the coefficient of proportionality. This is the only form of the control which assures the maintenance of a prescribed heading, and it can be used alone, or as a necessary component of a more sophisticated control.

The other forms of control functions affect the effective directional stability characteristics of a ship, but do not provide for the maintenance of a specified heading. These control functions are:

Velocity (or rate) control functions: $\delta=\beta\theta$
Acceleration control functions $\delta=\gamma\theta$ .....(14)
Integral control functions $\delta=\eta\int_t\theta(t)dt$

In addition there is usually a lag ε of the rudder motion with respect to the signal given by the change of the heading θ.

Complete expresions for the terms to be inserted on the right hand sides of equations (7) are

$$\begin{matrix} Y_\delta\,\delta\,(\alpha,\beta,\gamma,\eta,\varepsilon) \\ N_\delta\,\delta\,(\alpha,\beta,\gamma,\eta,\varepsilon) \end{matrix} \quad \ldots\ldots\ldots\ldots (15)$$

At the present status of technological development, the force Y_δ and moment N_δ can be evaluated only by model experiments.

MOTIONS OF A CONTROLLED SHIP IN WAVES

The equations of motions of a controlled ship in waves are obtained by combining equations 7, 13, and 15:

$$m_2\Psi'+\Psi Y_\psi-\theta'(m_1+Y_{\theta\cdot})=Y_w+Y_\delta\,\delta\,(\alpha,\beta,\gamma,\eta,\varepsilon)$$
$$I\theta''-\Psi N_\psi+\theta' N_{\theta\cdot}=N_w+N_\delta\,\delta\,(\alpha,\beta,\gamma,\eta,\varepsilon). \ldots\ldots (16)$$

Schiff and Gimprich [3] investigated the solutions of these equations for a ship in smooth water, replacing Y_w and N_w by random disturbances independent of ship hydrodynamics. Such disturbances can be caused, for instance, by wind gusts. Rydill [1] investigated the solution for Y_w and N_w caused by regular and irregular long-crested wave trains. The primary objectives of these investigations has been to determine the most advantageous set of rudder control characteristics, i.e. most advantageous values of α, β, γ and η, taking ε into account. The general conclusion was that proportional control, i.e. $\alpha\neq0$, $\beta=\gamma=\eta=0$, is satisfactory for a ship directionally stable in smooth water, i.e. characterized by a negative p_1. The addition of a rate control, i.e. $\beta>0$, $\gamma=\eta=0$, is needed for an unstable ship with positive p_1. It appears to be unnecessary to use the acceleration-proportional control and usually $\gamma=0$. Rydill pointed out that integral control, $\eta\neq0$, may lead to directional instability because of the 90 degrees lag it introduces. Therefore, the control problem essentially consists of a suitable choice of the coefficients of proportionality α and β.

Schiff and Gimrich demonstrated that the rudder lag ε is generally detrimental, but not critical in stable ships with negative p_1. It becomes seriously detrimental in unstable ships with positive p_1. Rydill, however, demonstrated the advantage of having a rudder phase lag which is a function of the frequency of wave encounter, $\varepsilon=\varepsilon(\omega_e)$.

In bow seas a ship experience wave disturbances at a high frequency and the combined displacement and rate automatic control causes excessive amount of rudder motion. Moreover, these rudder motions are useless because a ship cannot respond to them at high frequency. According to Rydill's analysis, there is hardly any difference in yawing motions of a normally stable ship in bow seas with and without rudder action. These useless rudder motions can be largely eliminated by providing a sufficient time lag ε at the high frequencies corresponding to bow seas.

On the other hand, at the low frequency of wave encounter in quartering sea both the wave disturbances and rudder forces have sufficient time to cause appreciable yawing motions of a ship. A minimum lag ε is desired in this case, so that wave disturbances can be promptly counteracted by the rudder. Such an adjustment of the time lag with frequency is usually observed in case of the manual steering and can be (in fact should be) incorporated in electronic devices used for automatic steering.

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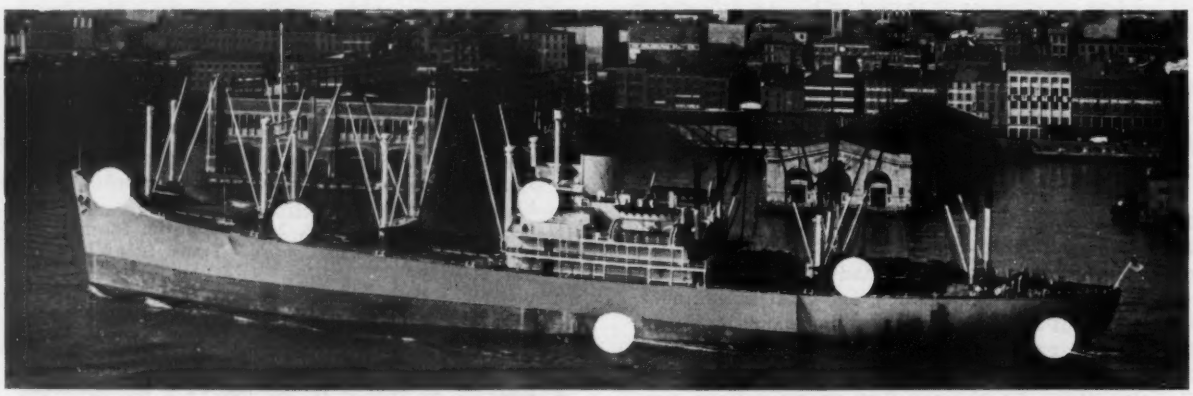

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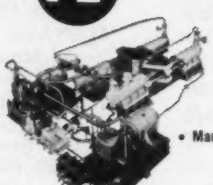

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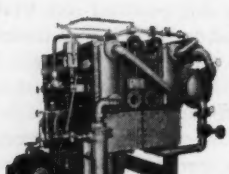

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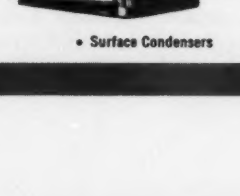

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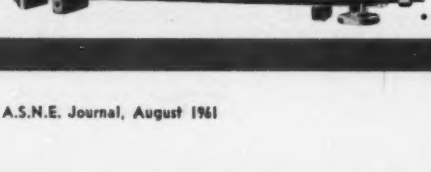

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THE ALIGNMENT OF GEAR CASES IN MANUFACTURE AND ON BOARD

ACKNOWLEDGMENT

This article is from the notes of the late G. Broersma. It appeared in the 1960 Annual Steam Number of "The Marine Engineer and Naval Architect" from which it is reprinted.

THE FIRST SYSTEM for alignment to be described, applies where the machining generally and boring of the bearings is carried out with the gear case mounted on steel rails laid in the floor of a large air-conditioned factory. It is assumed that Regnault-type boring machines are used and that these are mounted on heavy foundations, built up of cast iron cubes. The boring machines are aligned by means of water levels, needle gauges and plumb wires suspended from certain reference points. If a Berthiez type of milling and boring machine is used, then it may be assumed that bearing bores are produced which yield accurate parallelism and interdistance of the pinion and wheel shafting, i.e. the Berthiez machine itself is used for coordinating.

Whichever is the case, after machining in an air-conditioned shop, the basis for alignment checking is worked out as follows. Four reference points are welded to the gear case for future checking, these are arranged preferably at about the center height of the main wheel in a rectangular plan, if a square cannot be attained. This simplifies the subsequent

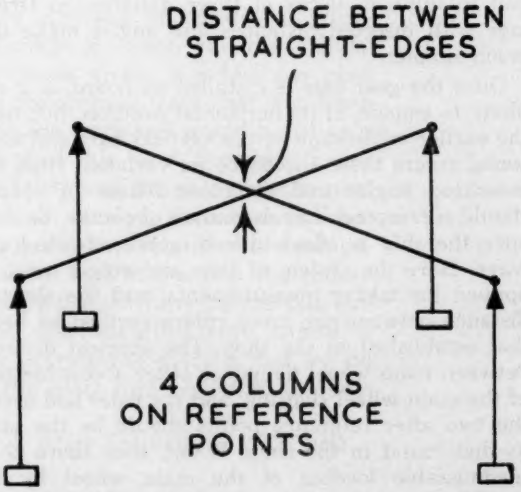


Figure 1. Measurement of vertical distance between crossed straight edges.

calculations. Vertical bars can then be erected on these reference points and two rulers laid diagonally on top of the vertical bars. (Figure 1). The bars, of course, should be of such a length that in the shop the rulers are horizontal. The actual position of the rulers is measured by means of adjustable water levels. The length of the vertical bars and the shortest vertical distance where the rulers cross are then measured. A cross-check on these measurements can be made by noting the relevant position of the two forward and the two after reference points, by making the two horizontal rulers parallel (Figure 2).

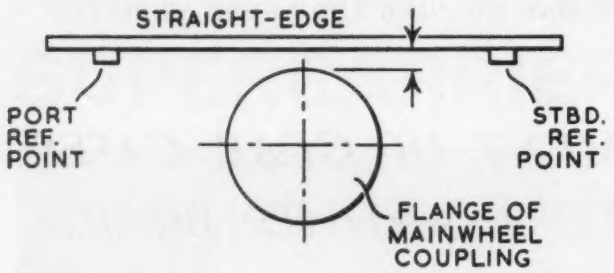


Figure 2. Cross checking by distance between reference places and coupling.

A ruler is also laid across the two after reference points and the shortest distance measured between the ruler and the main wheel flange, or other accessible part of the main shaft. These reference points should, of course, be over the center height of the main wheel. It is essential to make simple and clear drawings of the results of any measurements taken on a gearcase alignment, for it is difficult to think in terms of three dimensions. Drawings with inserted distances and angles make this much simpler.

Once the gear case is installed on board, it is unlikely to remain in its horizontal position, but from the earlier calibration of the vertical bars and horizontal rulers there should be no variation from the measured angles and distances. These of course, should correspond. Levels cannot, of course, be used once the ship is afloat, due to action of wind and wave. Here the system of bars and rulers must be applied for taking measurements, and the shortest distance between the cross rulers verified as being that established in the shop. The shortest distance between main wheel flange, or other accessible part of the main wheel shafting, and the ruler laid across the two after reference points should be the same as that found in the shop. If not, then there is an inadmissable loading of the main wheel by the propeller shaft. In this case it is clear that the gear case must be adjusted by suitable introduction of a large number of foundation chocks. This adjustment should be carried out in measured steps, by intermediate checking of the alignment and gradually tightening the foundation bolts all around. When the geometry is the same, within narrow limits, as in the gear shop, it is clear that the gearing has not been deformed unduly in aligning it aboard the vessel.

USE IN REPAIR WORK

In the case of repairs, say after re-cutting a wheel while the gear case remains in the vessel, it is necessary to make mandrels for both pinion and wheel bearings. It is, of course, assumed that the bearings are correct and this can be found by checking the feeler gauges at both ends of the bearing and with marking paint. The mandrels must now be checked for parallel with pin micrometers and adjustable water levels if the vessel is in dry dock. The position of the mandrels is checked with marking paint and feeler measurements at both ends of the bearing. After re-cutting the gears should be checked in the shops in a special rig. The pinion and wheel are first set at the correct centers and the backlash and bedding are checked. This bedding picture should be repeated when they are re-installed in the gear case on board the ship.

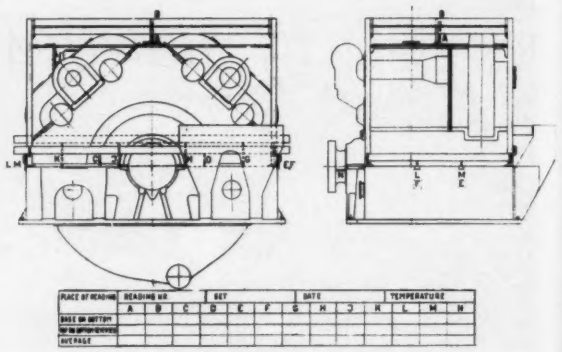


Figure 3. Four columns erected at the corners of a Maag gear case with straight edges in place and measuring points. A typical reference chart is given.

Alignment between the gear case and the shafting is performed by the conventional method of feelers with measuring blocks between the coupling mating flanges around the circumference. Alignment of repaired gears must be done with the utmost care, for it is not unusual to find that the shipbuilder's seatings are the cause of any difficulties which may have arisen. They are frequently insufficiently stiff, and may even work overnight. It is, of course, more effective to remove the whole gear case in case of trouble, and to carry out a real

job in the shop and at the same time to examine the seatings, which may well have been the original cause of trouble. Classification Societies and prudent owners will normally have the gear case removed, but occasionally distance may make this impossible. Should bearings have to be re-metalled, this must be performed first. It is usually desirable to weld reference points to the gear case if they do not already exist, before a major repair is undertaken.

MAAG PROCEDURE

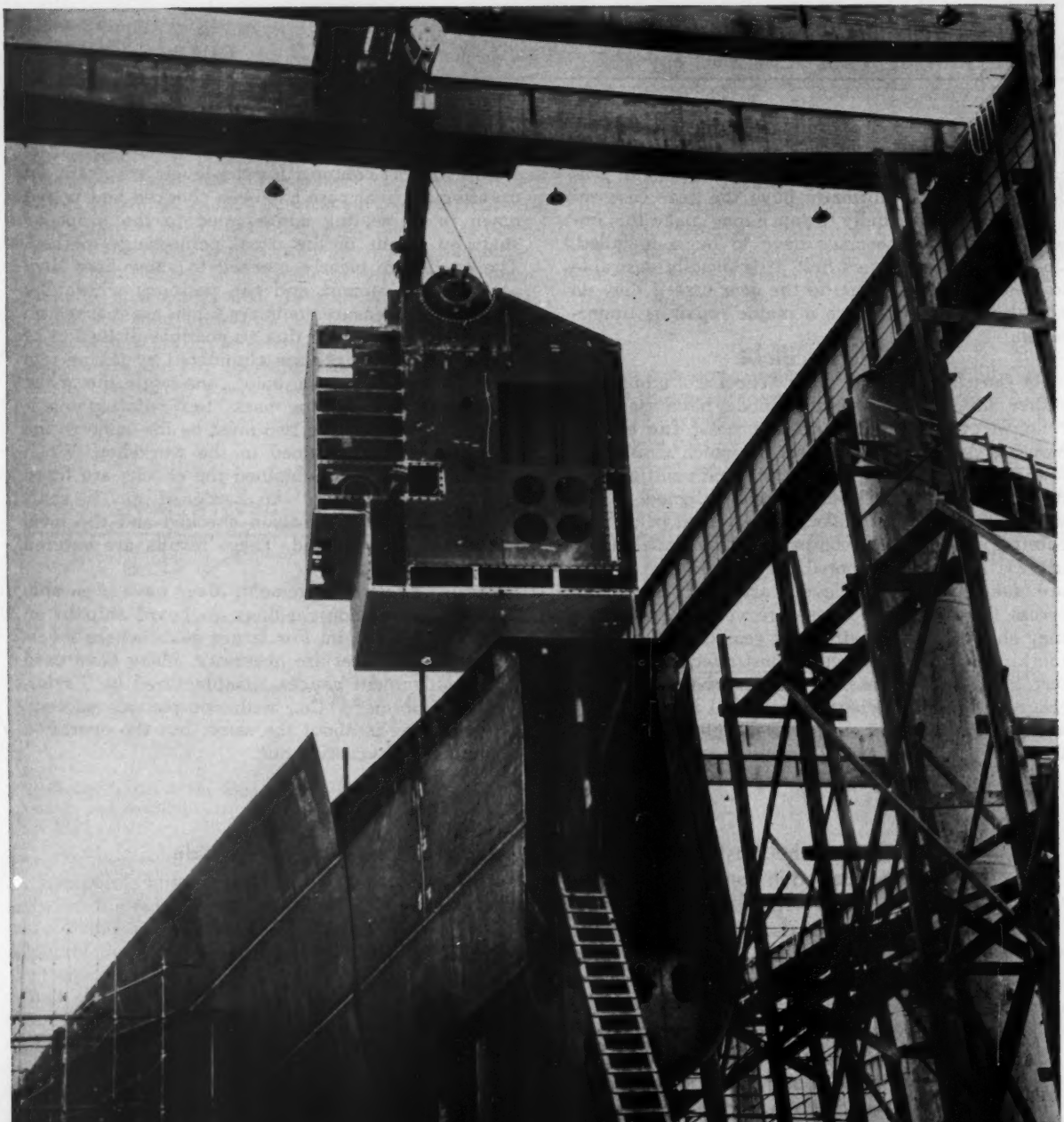
At Zurich, the Maag Gear Wheel Co. employing rather different alignment methods, have used mechanical gauges with marked success. The accompanying sketch shows the set-up which consists of four support columns, three levels and point gauges. After the gear has been properly aligned in the shop, the readings are taken as described below and entered in the form accompanying the sketch (Figure 3). It should be noted that readings A and B are taken with two levels arranged diagonally across the gear case. These are very important as they show clearly whether the gear case is twisted, the kind of distortion which most affects tooth contact. The levels themselves need not be of high precision and it is sufficient to use steel bars with top and bottom surfaces properly machined, provided that at each point measurements are taken as described. These readings must be repeated when the gear case is installed in the vessel.

The support columns for the levels are again set up after the gear case has been chocked and bolted down to its seating and aligned to the propeller shaft by means of the usual point-gauge method. The levels are clearly marked to show how they seat on the support and the positions where the point-gauge measurements are taken are marked on the gear case. Errors due to possible distortion of the levels themselves are eliminated by taking two measurements at each point, one with the mark "base" and one with the mark "top" pointing downwards. Average of the two must be the same as the original average obtained in the workshop. When correct alignment is obtained the chocks are fitted and the holding-down bolts hardened up. The shaft coupling alignment is again checked and the level measurements repeated. These results are entered on the record sheet.

By using this arrangement, Maag have been able to repeat workshop readings on board ship to an accuracy of 0.001 in. For larger gears where levels longer than 10 feet are necessary, Maag have used optical alignment gauges, manufactured by Taylor, Taylor, Hobson, & Co., with conspicuous success. The accuracy is about the same, but the operation is more readily carried out.

One of the most common components in a ship's structure is a wide plate loaded in compression along the long edge. Deck and bottom plates in a transversely framed ship are obvious examples. Yet there has been a shortage of data on this specific matter as the work done by Dr. J. Montgomerie before the war was based on a different type of hull structure from that now built. A paper presented to the Society of Naval Architects and Marine Engineers by Professor Harvey J. Evans describes some work done at the Massachusetts Institute of Technology in recent years, the tests covering 192 plates with three degrees of rotational edge-restraint along the loaded edges. A formula is given for the critical buckling stress and it is said that the ultimate stress is a few per cent higher. The formula agrees well with the theoretical work that has been done and is in a form suitable for use in design work and would be of help to anyone who had to deal with a problem of this type. Professor Evans says that the test conditions represented fairly well the sort of unfairness that would occur in an ordinary ship plate in service, even though from the tests it was not possible to obtain a quantitative relation between unfairness and the buckling load. It would seem that in the ship the redundant nature of the structure would probably prevent any dangerous collapse. A number of years back the classification societies were forced to take action in regard to ships with badly corrugated plating, but we do not remember ever seeing one in which the plate had collapsed under a compressive loading.

—SHIPBUILDING AND SHIPPING RECORD



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N. GASPAROVIC

THE TRANSMISSION RATIO OF DOUBLE-REDUCTION GEARING FOR SHIP PROPULSION*

THE AUTHOR

obtained a degree of Marine Engineering at the University of Zagreb. For three years he dealt with ship's propulsion machinery and standardization of ship's equipment. He now belongs to the editorial staff of the journal "Brennstoff-Warme-Kraft." He has published several articles on gas turbines and ship's propulsion systems.

INTRODUCTION

Gear drives are frequently used on ships. With turbine propulsion gearing is essential to reduce the high speed of the turbine to the low speed of the propeller. The reduction ratio is usually between 35 and 55, which is too high for a single gear train and double-reduction gearing must therefore be employed.

The weight of such gearing often exceeds that of the turbine, so that obviously every effort must be made to reduce the weight. Choice of suitable materials and appropriate design are important in this respect but with two-stage and multi-stage gearing the speed of the lay shaft is the major factor. With unsuitable division of the reduction ratio between the first and second stages all the improvements otherwise achieved are unavailing. It must also not be overlooked that lower weight means lower costs of material and machining.

The purpose of this study is to determine the speed of the lay shaft which gives the lowest weight of the gearwheels and thus essentially of the complete drive. The results show that in modern practice the reduction ratios selected for the second stage are too high. In this sense significant changes in future designs are necessary.

DIMENSIONS OF THE PINION

With a gear drive the conditions as regards strength of material are only critical in the case of the pinion. As a starting point the permissible stress σ_{zul} at the root of the tooth may be taken as a reference [1].

$$\sigma_{zul}=qP/bm \quad \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

in which q is a coefficient depending on the number of teeth, the pressure angle and the profile offset, P is the tangential force, b is the width and m the modulus. Further

$$m=d/z \quad \ldots\ldots\ldots\ldots\ldots\ldots (2)$$

with d as pitch diameter and z the number of teeth.

If the power is designated by N and the speed by n, the tangential force is

$$P=N/d\,\pi\,n. \quad \ldots\ldots\ldots\ldots\ldots\ldots (3)$$

If equations (2) and (3) are inserted into (1) we get

$$\sigma_{zul}=qzN/d^2b\,\pi\,n \quad \ldots\ldots\ldots\ldots\ldots\ldots (4)$$

* Translated from "Schiffstechnik," 1(1960)35, pp. 27/31.

and thus the dimensions of the pinion

$$d^2b=qzN/\pi\, n\, \sigma_{zul}\,. \quad \ldots\ldots\ldots\ldots\ldots (5)$$

With helical gears the force at the root of the tooth is 2/3≈0.67 less than with straight-toothed gears and with β as helix angle equation (5) becomes

$$d^2b=0.67\, qzN/\pi\, n\, \sigma_{zul} \cos\beta\,. \quad \ldots\ldots\ldots\ldots (6)$$

For ship propulsion double helical gears are used, the side thrust of one half of the wheel being cancelled by that of the other. The width b in equations refers to the active face width of the two halves of the wheel.

ONE DRIVING SHAFT AND ONE LAY SHAFT

As will be seen from Figures 1 and 2 a double-reduction gear consisting of one driving shaft and one lay shaft can be arranged in two ways

The total weight of the gears is

$$G=G_1+G_2+G_3+G_4, \quad \ldots\ldots\ldots\ldots\ldots (7)$$

in which G_1 to G_4 are the weights of the wheels and pinions 1 to 4.

If the specific gravity γ of the material for all the wheels is the same, their weights are:—

$$G_1=d_1^2b_1\,\pi\gamma/4 \quad \ldots\ldots\ldots\ldots\ldots (8)$$
$$G_2=\varepsilon\, d_2^2b_2\,\pi\gamma/4. \quad \ldots\ldots\ldots\ldots\ldots (9)$$
$$G_3=d_3^2b_3\,\pi\gamma/4 \quad \ldots\ldots\ldots\ldots\ldots (10)$$
$$G_4=\varepsilon\, d_4^2b_4\,\pi\gamma/4 \quad \ldots\ldots\ldots\ldots\ldots (11)$$

in which d_1 to d_4 are the corresponding pitch diameters and b_1 to b_4 the widths. As wheels 2 and 4 are not constructed as solid cylinders a factor ε is included in equations (9) and (11) to take account of this. ε depends on the diameter, the material used and on the design of the wheel; the average value can be taken as 0.2 to 0.3. The same factor ε was assumed in equations (9) and (11). As will be seen later it has a negligible influence on the speed of the lay shaft. The diameters of wheels 2 and 4 are derived from

$$d_2=d_1\, n/n' \quad \ldots\ldots\ldots\ldots\ldots (12)$$

and

$$d_4=d_3\, n'/n_o \quad \ldots\ldots\ldots\ldots\ldots (13)$$

For the widths we take

$$\left.\begin{aligned} b_2&=b_1\\ b_4&=b_3 \end{aligned}\right\} \quad \ldots\ldots\ldots\ldots\ldots (14)$$

Equations (9) and (11) accordingly become

$$G_2=\varepsilon\, d_1^2b_1\,(n/n')^2\,\pi\gamma/4 \quad \ldots\ldots\ldots\ldots (15)$$
$$G_4=\varepsilon\, d_3^2b_3\,(n'/n_o)^2\,\pi\gamma/4 \quad \ldots\ldots\ldots\ldots (16)$$

In view of equations (5) and (6) respectively the relationship between the dimensions of the pinions 1 and 3, assuming the same values for q, z, N, σ_{zul} and β, is:

$$d_3^2b_3=d_1^2b_1\, n/n'. \quad \ldots\ldots\ldots\ldots\ldots (17)$$

Equations (10) and (11) accordingly now become

$$G_3=d_1^2b_1\,(n/n')\,\pi\gamma/4 \quad \ldots\ldots\ldots\ldots\ldots (18)$$
$$G_4=\varepsilon\, d_1^2b_1\,(nn'/n_o^2)\,\pi\gamma/4 \quad \ldots\ldots\ldots\ldots (19)$$

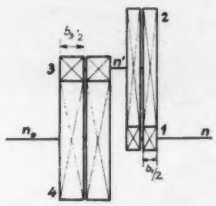

Figure 1. Two-stage gear drive with first stage outside. n speed of driving shaft, n′ speed of lay shaft, n_o output speed.

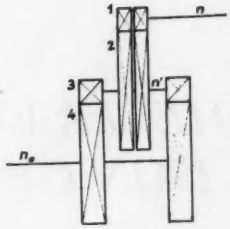

Figure 2. Two-stage gear with first stage inside.

The total weight of the gears is thus

$$G=[1+\varepsilon(n/n')^2+(n/n')+(\varepsilon\, nn'/n_o^2)]d_1^2b_1\,\pi\gamma/4\,. \quad \ldots\ldots\ldots\ldots\ldots (20)$$

If now equation (20) is differentiated with respect to the speed n′ of the lay shaft and the result is equated to zero, i.e.

$$\partial G/\partial n'=0, \quad \ldots\ldots\ldots\ldots\ldots (21)$$

we get an equation of the third degree

$$n'^3-(n_o^2/\varepsilon)n'-2\, nn_o^2=0. \quad \ldots\ldots\ldots\ldots (22)$$

As we are concerned with a minimum value of the relationship $G=f(n')$, it is obvious that $\partial^2G/\partial n'^2>0$. It follows from equation (20) that

$$\partial^2G/\partial n'^2=[3n'^2-(n_o^2/\varepsilon)]\; d_1^2b_1\,\pi\gamma/4>0, \quad \ldots\ldots (23)$$

which proves that there is in fact a minimum value for the weight. With the transmission ratio i of the gear

$$i=n/n_o \quad \ldots\ldots\ldots\ldots\ldots (24)$$

we can now transform equation (22) as follows:

$$n'^3-(n_o^2/\varepsilon)n'-2\, in_o^3=0 \quad \ldots\ldots\ldots\ldots (25)$$

There is only one real root of equation (25) because the discriminant

$$i^2n_o^6-(n_o^6/27\,\varepsilon^3)=i^2n_o^6[1-(1/27\,\varepsilon^3i^2)\;]>0 \quad \ldots. (26)$$

The solution is

$$n'=\sqrt[3]{in_o^3+\sqrt{i^2n_o^6[1-(1/27\,\varepsilon^3i^2)\,]}} \quad \ldots\ldots\ldots (27)$$
$$+\sqrt[3]{in_o^3-\sqrt{i^2n_o^6[1-(1/27\,\varepsilon^3i^2)\,]}}$$

from which it follows that

$$n' = i^{1/3} n_0 \{ \sqrt[3]{1+\sqrt{1-(1/27)\,\varepsilon^3 i^2}} + \sqrt[3]{1-\sqrt{1-(1/27\,\varepsilon^3 i^2)}} \} \quad \ldots\ldots (28)$$

If $1/27\,\varepsilon^3\,i^2$ is equated to zero, the expression in the braces { } is to a very good approximation equal to $\sqrt[3]{2} \approx 1.26$ and equation (28) becomes

$$n' = 1.26\, i^{1/3} n_0 \quad \ldots\ldots (29)$$

With the transmission ratio i' of the lay shaft

$$i' = n'/n_0 \quad \ldots\ldots (30)$$

i.e. the transmission ratio of the second stage, we get finally

$$i' = 1.26\, i^{1/3} \quad \ldots\ldots (31)$$

This result with the difference due to the approximation $1/27\,\varepsilon^3\,i^2 = 0$ as compared with $1/27\,i^2 = 0$ is equivalent to the result in which also the weights of wheels 2 and 4 were assumed as those of solid cylinders [2].

The relationship according to equation (31) is shown in Figure 3.

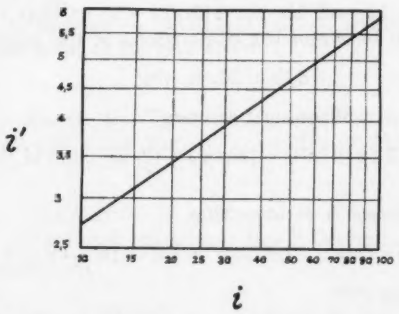

Figure 3. Transmission ratio of the lay shaft related to the transmission ratio of the gear drive. ((In the diagram)) Transmission ratio of the second stage i'. Transmission ratio i.

With the arrangement as in Figure 2 with the first stage inside, because of the distance between the lay shaft and the output shaft the following condition must be fulfilled:

$$d_2 < d_3 + d_4 \quad \ldots\ldots (32)$$

With equations (12) and (13) it further follows that

$$d_1/d_3 < [1+(n'/n_0)]\,n'/n \quad \ldots\ldots (33)$$

From equation (17) we obtain

$$d_1/d_3 = (b_3 n'/b_1 n)^{1/2} \quad \ldots\ldots (34)$$

which together with equation (29) inserted into equation (33) for the optimum speed of the lay shaft gives

$$b_3/b_1 < 1.26\, i^{-2/3}(1+1.26\, i^{1/3})^2 \quad \ldots\ldots (35)$$

This relationship is shown in Figure 4

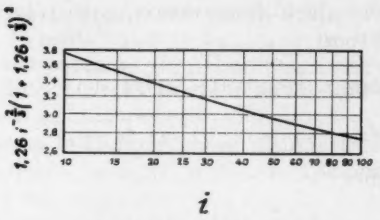

Figure 4. Ratio of the widths of pinions related to the transmission ratio of the gear drive.

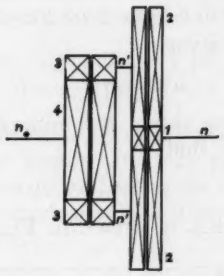

Figure 5. Two-stage gear drive with one driving shaft and two lay shafts.

ONE DRIVING SHAFT AND TWO LAY SHAFTS

Figure 5 shows schematically a gear drive with one driving shaft and two lay shafts.

The total weight of this drive is

$$G = G_1 + 2G_2 + 2G_3 + G_4 \quad \ldots\ldots (36)$$

in which for G_1 to G_4 equations (8), (15), (10) and (16) are applicable:

$$G = \{[1+2\,\varepsilon(n/n')^2]\,d_1^2 b_1 + [2+\varepsilon(n'/n_0)^2]\,d_3^2 b_3\}\,\pi\gamma/4 \quad \ldots\ldots (37)$$

As, however, only half the power acts on the pinions 3, i.e. N/2, the following expression arises instead of equation (17)

$$d_3 b_3 = d_1^2 b_1\, n/2n' \quad \ldots\ldots (38)$$

and (37) accordingly becomes

$$G = [1+2\,\varepsilon(n/n')^2 + (n/n') + (\varepsilon\, nn'/2n_0^2)]\,d_1^2 b_1 \pi\gamma/4 \quad \ldots\ldots (39)$$

If now the same procedure $\partial G/\partial n' = 0$ is followed, we again get an equation of the third degree:

$$n'^3 - (2n_0^2/\varepsilon)\,n' - 8nn_0^2 = n'^3 - (2n_0^2/\varepsilon)\,n' - 8in_0^3 = 0 \quad \ldots\ldots (40)$$

Further

$$\partial^2 G/\partial n'^2 = [3n'^2 - (2n_0^2/\varepsilon)]\,d_1^2 b_1 \pi\gamma/4 > 0 \quad \ldots\ldots (41)$$

The discriminant

$$16 i^2 n_0^6 - (8n_0^6/27\,\varepsilon^3) = 16 i^2 n_0^6 [1 - (1/54\varepsilon^3 i^2)] > 0 \quad \ldots\ldots (42)$$

is positive, which shows that equation (40) has only one real root:

$$n' = \sqrt[3]{4in_0^3 + \sqrt{16i^2n_0^6[1-(1/54\,\varepsilon^3i^2)]}} + \sqrt[3]{4in_0^3 - \sqrt{16\,i^2n_0^6[1-(1/54\,\varepsilon^3i^2)]}} \quad \ldots\ldots (43)$$

and further

$$n' = i^{1/3}n_0\{\sqrt[3]{4+\sqrt{16[1-(1/54\,\varepsilon^3i^2)]}} + \sqrt[3]{4-\sqrt{16[1-(1/54\,\varepsilon^3i^2)]}}\} \quad \ldots\ldots (44)$$

If now $1/54\,\varepsilon^3 i^2$ is equated to zero, the expression in the braces is to a very good approximation equal to $\sqrt[3]{8}=2$. This gives:

$$n' = 2i^{1/3}n_0 \quad \ldots\ldots (45)$$

or with (30) we obtain the optimum transmission ratio of the lay shaft

$$i' = 2i^{1/3} \quad \ldots\ldots (46)$$

which relationship is shown in Figure 6.

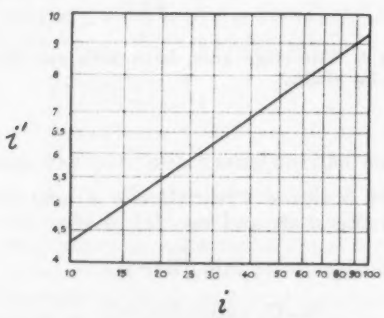

Figure 6. Transmission ratio of the lay shaft related to the transmission ratio of the gear drive ((In the diagram)). Transmission ratio of the second stage i'. Transmission ratio i.

If the weights of the gears as given by equations (20) and (39) are compared, the appropriate optimum speed n' according to equation (29) being inserted in (20) and that from equation (45) in (39), it is seen to be advantageous to construct the gear system with two lay shafts. In this study the weight of the casing was not considered and this might be very important. The weight of the casing and also the manufacturing costs are higher for gear drives with two lay shafts.

TWO DRIVING SHAFTS AND ONE LAY SHAFT

Figure 7 shows schematically a gear drive with two driving shafts and one lay shaft. The case is first considered in which for the driving shafts the following conditions for the power and speed are fulfilled:

$$\left.\begin{aligned} N_a &= N_b = N/2 \\ n_a &= n_b = n \end{aligned}\right\} \quad \ldots\ldots (47)$$

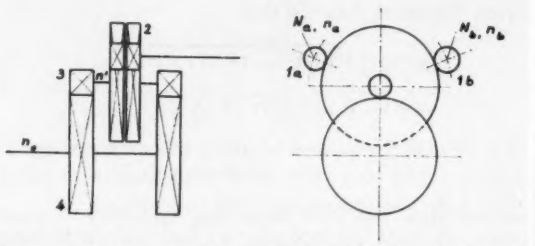

Figure 7. Two-stage drive with two driving shafts and one lay shaft.

from which it follows that the weight of the pinions is

$$G_{1a} = G_{1b} = G_1 \quad \ldots\ldots (48)$$

The total weight of the gears is accordingly

$$G = 2G_1 + G_2 + G_3 + G_4 \quad \ldots\ldots (49)$$

and making use of equations (8), (15), (10) and (16):

$$G = \{[2+\varepsilon(n/n')^2]d_1^2b_1 + [1+\varepsilon(n'/n_0)^2]d_3^2b_3\}\pi\gamma/4 \quad \ldots\ldots (50)$$

The power acting on pinion 3 is double that on pinions 1a and 1b, i.e. $2(N/2)=N$, so that the relationship between the dimensions of the pinions is

$$d_3^2b_3 = 2d_1^2b_1\,n/n' \quad \ldots\ldots (51)$$

Equation (50) now becomes

$$G = [2+\varepsilon(n/n')^2 + (2n/n') + (2\varepsilon\,nn'/n_0^2)]d_1^2b_1\pi\gamma/4 \quad \ldots\ldots (52)$$

By analogy it follows that

$$n'^3 - (n_0^2/\varepsilon)\,n' - nn_0^2 = n'^3 - (n_0^2/\varepsilon)\,n' - in_0^3 = 0 \quad \ldots\ldots (53)$$

with the root

$$\begin{aligned} n' &= \sqrt[3]{(in_0^3/2) + \sqrt{[1-(4/27\,\varepsilon^3i^2)]i^2n_0^6/4}} + \sqrt[3]{(in_0^3/2) - \sqrt{[1-(4/27\,\varepsilon^3i^2)]i^2n_0^6/4}} \\ &= i^{1/3}n_0\{\sqrt[3]{0.5+\sqrt{[1-(4/27\,\varepsilon^3i^2)]/4}} + \sqrt[3]{0.5-\sqrt{1-(4/27\,\varepsilon^3i^2)/4}}\} \end{aligned} \quad \ldots\ldots (54)$$

Assuming that $4/27\,\varepsilon^3\,i^2$ is negligibly small, the expression in the braces { } is equal to $\sqrt[3]{1}=1$, i.e.

$$n' = i^{1/3}n_0 \quad \ldots\ldots (55)$$

or

$$i' = i^{1/3} \quad \ldots\ldots (56)$$

which relationship is shown in Figure 8.

As in the case of the arrangement in Figure 2 the condition (32) must be fulfilled. From (51)

$$d_1/d_3 = (b_3n'/2b_1n)^{1/2} \quad \ldots\ldots (57)$$

and together with equation (55) for the optimum speed inserted into (33)

$$b_3/b_1 < 2i^{-2/3}[1+i^{1/3}]^2 \quad \ldots\ldots (58)$$

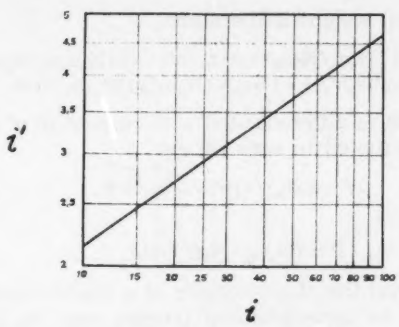

Figure 8. Transmission ratio of the lay shaft related to the transmission ratio of the gear drive. ((In the diagram)). Transmission ratio of the second stage i'. Transmission ratio i.

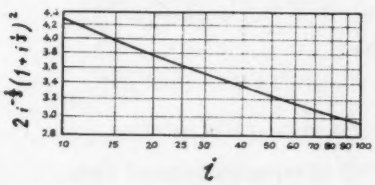

Figure 9. Ratio of the widths of pinions related to the transmission ratio of the gear drive.

which is shown in Figure 9. If the first stage is placed outside as in Figure 1, condition (32) does not apply.

The same gear drive as is shown in Figure 7 will now be considered, to which the following is generally applicable:

$$N_a \neq N_b$$
$$N_a + N_b = N \quad \ldots\ldots (59)$$
$$n_a \neq n_b$$

As regards Figure 7 as the pinions 1a and 1b engage with a common wheel 2, the following further conditions must be fulfilled:

$$d_{1a} n_{1a} = d_{1b} n_b \quad \ldots\ldots (60)$$

and

$$z_{1a}/z_{1b} = d_{1a}/d_{1b} \quad \ldots\ldots (61)$$

We further assume that the two pinions are of the same width

$$b_{1a} = b_{1b} = b_1 \quad \ldots\ldots (62)$$

in which the width b_1 must be based on the width of the "critical" pinion For this we constitute the relationship

$$d^2_{1a} b_{1a} / d^2_{1b} b_{1b} = q_{1a} z_{1a} N_a n_b / q_{1b} z_{1b} N_b n_a \quad \ldots\ldots (63)$$

which with (60) and (61) gives

$$b_{1a}/b_{1b} = q_{1a} N_a / q_{1b} N_b \quad \ldots\ldots (64)$$

If now the expression on the right side of (64) is greater than 1, b_1 is made equal to b_{1a} and if less than 1, b_1 is made equal to b_{1b}, b_{1a} or b_{1b} being calculated from equation (5). The other "non-critical" pinion is then made of the same width as the "critical" one in accordance with (62).

For further investigation we assume that pinion 1a is "critical" i.e. $q_{1a} N_a / q_{1b} N_b > 1$. The individual weights of the gears are:

$$G_{1a} = d^2_{1a} b_1 \pi\gamma/4 \quad \ldots\ldots (65)$$
$$G_{1b} = d^2_{1b} b_1 \pi\gamma/4 = d^2_{1a} b_1 (n_a/n_b)^2 \pi\gamma/4 \quad \ldots\ldots (66)$$
$$G_2 = \varepsilon\, d^2_{1a} b_1 (n_a/n')^2 \pi\gamma/4 \quad \ldots\ldots (67)$$

For gears 3 and 4 equations (10) and (16) apply. The total weight is accordingly:

$$G = \{[1 + (n_a/n_b)^2 + \varepsilon (n_a/n')^2] d^2_{1a} b_1 + [1 + \varepsilon (n'/n_0)^2] d^2_3 b_3\} \pi\gamma/4 \quad \ldots\ldots (68)$$

As regards the dimensions of the pinion 3 there are two possibilities: either $z_3 = z_{1a}$ or $z_3 = z_{1b}$ according to whether pinion 1a or 1b has the smaller number of teeth, which number will also be chosen for pinion 3. When $z_3 = z_{1a}$

$$d^2_3 b_3 = d^2_{1a} b_1 N n_a / N_a n' \quad \ldots\ldots (69)$$

and when $z_3 = z_{1b}$

$$d^2_3 b_3 = d^2_{1b} b_1 N n_b / N_b n' = d^2_{1a} b_1 N n^2_a / N_b n_b n' \quad \ldots\ldots (70)$$

or in general

$$d^2_3 b_3 = 2 d^2_{1a} b_1 n^2_a / i^* n_0 n' \quad \ldots\ldots (71)$$

where

$$\left.\begin{array}{l} \text{for } z_3 = z_{1a}\text{: } i^* = 2N_a n_a / N n_0 \\ \text{for } z_3 = z_{1b}\text{: } i^* = 2N_b n_b / N n_0 \end{array}\right\} \quad \ldots\ldots (72)$$

which quantity has the characteristic of a transmission ratio related to corresponding powers.

Equation (68) for the total weight now becomes:

$$G = [1 + (n_a/n_b)^2 + \varepsilon (n_a/n')^2 + (2n^2_a/i^* n_0 n') + (2\varepsilon\, n^2_a n'/i^* n^3_0)] d^2_{1a} b_1 \pi\gamma/4 \quad \ldots\ldots (73)$$

By means of the procedure $\partial G/\partial n' = 0$ it follows that

$$n'^3 - (n^2_0/\varepsilon) n' - i^* n^3_0 = 0 \quad \ldots\ldots (74)$$

which expression when $n_a = n_b$ and $N_a = N_b = N/2$ is equivalent to (53).

If now equations (74) and (53) are compared it is seen that they are of the same form and the transmission ratio i* has appeared instead of i. The solution is also similar

$$n' = i^{*\frac{1}{3}} n_0 \quad \ldots\ldots (75)$$

or

$$i' = i^{*\frac{1}{4}} \quad \ldots\ldots (76)$$

which relationship is shown in Figure 8, where i* must be inserted instead of i.

The question is now which i* according to (72) must be used: If $n_a > n_b$ then $z_3 = z_{1a}$ and conversely if $n_b > n_a$ then $z_3 = z_{1b}$ with in each case the corresponding value of i*.

DOUBLE-REDUCTION GEARS

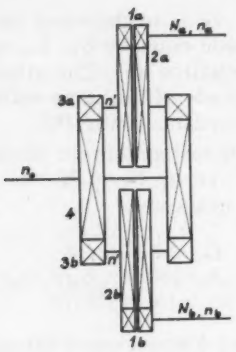

Figure 10. Two-stage gear drive with two driving shafts and two lay shafts.

TWO DRIVING SHAFTS AND TWO LAY SHAFTS

Figure 10 shows schematically a two-stage gear drive with two driving shafts and two lay shafts, it being assumed that the lay shafts run at the same speed n'.

This case will be considered quite generally from the following aspects:

$$\left.\begin{array}{l} N_a > N_b \\ N_a + N_b = N \\ n_a \neq n_b \end{array}\right\} \quad \ldots\ldots (77)$$

the speeds of the two lay shafts being the same. It follows that

$$d_{2a}/d_{1a} = n_a/n' \quad \ldots\ldots (78)$$

and

$$d_{2b}/d_{1b} = n_b/n' \quad \ldots\ldots (79)$$

The numbers of teeth z_{1a} and z_{1b} of pinions 1a and 1b are also assumed to be equal, which because of (6) gives

$$d^2_{1b} b_{1b} = d^2_{1a} b_{1a}\, N_b n_a / N_a n_b \quad \ldots\ldots (80)$$

The width b_3 of pinions 3a and 3b is taken as the same so that since $N_a > N_b$ pinion 3a is "critical." Accordingly since $G_{3a} = G_{3b} = G_3$ the total weight

$$G = G_{1a} + G_{1b} + G_{2a} + G_{2b} + 2G_3 + G_4 \quad \ldots\ldots (81)$$

the individual weights being

$$G_{1a} = d^2_{1a} b_{1a} \pi\gamma/4 \quad \ldots\ldots (82)$$

$$G_{1b} = d^2_{1b} b_{1b} \pi\gamma/4 = d^2_{1a} b_{1a} (N_b n_a / N_a n_b) \pi\gamma/4 \quad \ldots\ldots (83)$$

$$G_{2a} = \varepsilon\, d^2_{2a} b_{2a} \pi\gamma/4 = \varepsilon\, d^2_{1a}\, b_{1a} (n_a/n')^2 \pi\gamma/4 \quad \ldots\ldots (84)$$

$$\left.\begin{array}{l} G_{2b} = \varepsilon\, d^2_{2b} b_{2b} \pi\gamma/4 = \varepsilon\, d^2_{1b}\, b_{1b} (n_b/n')^2 \pi\gamma/4 \\ \quad = \varepsilon\, d^2_{1a} b_{1a} (N_b n_a n_b / N_a n'^2) \pi\gamma/4 \end{array}\right\} \quad \ldots\ldots (85)$$

for G_3 and G_4 equations (10) and (16) apply which with

$$d^2_3 b_3 = d^2_{1a} b_{1a} N n_a / N_a n' \quad \ldots\ldots (86)$$

gives

$$G_3 = d^2_{1a} b_{1a} (N n_a / N_a n') \pi\gamma/4 \quad \ldots\ldots (87)$$

$$G_4 = \varepsilon\, d^2_{1a} b_{1a} (N n_a n' / N_a n_o^2) \pi\gamma/4 \quad \ldots\ldots (88)$$

The total weight is therefore

$$G = [1 + (N_b n_a / N_a n_b) + \varepsilon (n_a/n')^2 + \varepsilon (N_b n_a n_b / N_a n'^2) + 2(N n_a / N_a n') + \varepsilon (N n_a n' / N_a n_o^2)]\, d^2_{1a} b_{1a} \pi\gamma/4 \quad \ldots\ldots (89)$$

If (89) is differentiated with respect to n' and the result equated to zero we get

$$n'^3 - (2n_o^2/\varepsilon) n' - 2i^* n_o^3 = 0, \quad \ldots\ldots (90)$$

where

$$i^* = (N_a n_a + N_b n_b)/N n_o, \quad \ldots\ldots (91)$$

which has the characteristic of a transmission ratio related to corresponding powers (not to be confused with i* from equation (72) !). When $N_a = N_b$ and $n_a = n_b = n$, $i^* = i$.

The solution of equation (90) gives

$$n' = \sqrt[3]{i^* n_o^3 + \sqrt{i^{*2} n_o^6 [1 - (8/27\, \varepsilon^3 i^{*2})]}} + \sqrt[3]{i^* n_o^3 - \sqrt{i^{*2} n_o^6 [1 - (8/27\, \varepsilon^3 i^{*2})]}} \quad \ldots\ldots (92)$$

and further

$$n' = i^{*\frac{1}{3}} n_o \{ \sqrt[3]{1 + \sqrt{1 - (8/27\, \varepsilon^3 i^{*2})}} + \sqrt[3]{1 - \sqrt{1 - (8/27\, \varepsilon^3 i^{*2})}} \} \quad \ldots\ldots (93)$$

If $8/27\, \varepsilon^3\, i^{*2}$ is equated to zero, then

$$n' = 1.26\, i^{*\frac{1}{3}} n_o \quad \ldots\ldots (94)$$

and finally

$$i' = 1.26\, i^{*\frac{1}{3}} \quad \ldots\ldots (95)$$

This relationship is shown in Figure 3 but the transmission ratio i* related to the corresponding powers must be taken instead of i.

It can be shown that the gear drive as in Figure 10 with two lay shafts is considerably heavier than that as in Figure 7 with one lay shaft. The casing is also heavier and therefore gearing with two driving shafts and two lay shafts should never be used.

THREE DRIVING SHAFTS AND TWO LAY SHAFTS

Finally the case will be considered where three different powers N_a, N_b and N_c with three different speeds "feed" the gear drive. As shown in Figure 11 the pinions 1a and 1b on the driving shafts N_a, n_a and N_b, n_b engage with a common wheel.

The following assumptions are made:

$$\left.\begin{array}{l} N_a + N_b \geqq N_c \\ N_a + N_b + N_c = N \end{array}\right\} \quad \ldots\ldots (96)$$

from which it follows that the pinions 3a and 3c, which are assumed to be identical ($G_{3a} = G_{3c} = G_3$) and run at the same speed n' are designed for the power $N_a + N_b$.

Conditions (60) and (61) must also be fulfilled, according to which the pinions should be of the same width $b_{1a} = b_{1b} = b_1$ [condition (62)]. It is assumed in accordance with (63) to (66) that pinion 1a is "critical" with respect to 1b, which further means that all weights must be related to the pinion 1a.

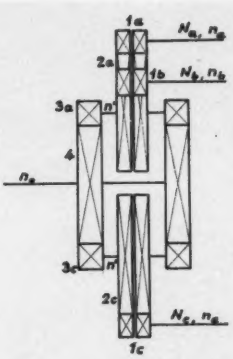

Figure 11. Two-stage gear drive with three driving shafts and two lay shafts.

The following also apply,

$$\left.\begin{aligned} d_{2a}/d_{1a} &= n_a/n' \\ d_{2b}/d_{1b} &= n_b/n' \\ d_{2c}/d_{1c} &= n_c/n \end{aligned}\right\} \quad \ldots\ldots\ldots\ldots\ldots\ldots (97)$$

The total weight is

$$G = G_{1a}+G_{1b}+G_{1c}+G_{2a}+G_{2c}+2G_3+G_4 \quad \ldots\ldots (98)$$

For G_{1a}, G_{1b} and G_{2a} equations (65) to (67) apply. For the pinion 1c we have

$$G_{1c} = d_{1c}^2 b_{1c} \pi\gamma/4 \quad \ldots\ldots\ldots\ldots\ldots (99)$$

If now $z_{1c}=z_{1a}$ it follows because of (6) that

$$d_{1c}^2 b_{1c} = d_{1a}^2 b_1 N_c n_a / N_a n_c \quad \ldots\ldots\ldots\ldots\ldots (100)$$

or if $z_{1c}=z_{1b}$

$$d_{1c}^2 b_{1c} = d_{1b}^2 b_1 N_c n_b / N_b n_c = d_{1a}^2 b_1 N_c n_a^2 / N_b n_b n_c \quad \ldots\ldots (101)$$

For simplicity we write equations (100) and (101) in generalized form

$$d_{1c}^2 b_{1c} = d_{1a}^2 b_1 \mu_1 \quad \ldots\ldots\ldots\ldots\ldots (102)$$

where

$$\begin{aligned} &\text{for } z_{1c}=z_{1a}: \ \mu_1 = N_c n_a / N_a n_c \\ &\text{for } z_{1c}=z_{1b}: \ \mu_1 = N_c n_a^2 / N_b n_b n_c . \end{aligned} \quad \ldots\ldots\ldots\ldots (103)$$

Accordingly

$$G_{1c} = d_{1c}^2 b_{1c} \pi\gamma/4 = d_{1a}^2 b_1 \mu_1 \pi\gamma/4 \quad \ldots\ldots (104)$$

and

$$G_{2c} = \varepsilon\, d_{2c}^2 b_{2c} \pi\gamma/4 = \varepsilon\, d_{1a}^2 b_1 (n_c/n')^2 \mu_1 \pi\gamma/4 \quad \ldots (105)$$

By analogy with (69) and (70) for $z_3=z_{1a}$:

$$d_3^2 b_3 = d_{1a}^2 b_1 (N_a+N_b) n_a / N_a n' \quad \ldots\ldots\ldots (106)$$

or for $z_3=z_{1b}$:

$$\begin{aligned} d_3^2 b_3 &= d_1^2 b_1 (N_a+N_b) n_b / N_b n' \\ &= d_{1a}^2 b_1 (N_a+N_b) n_a^2 / N_b n_b n' \end{aligned} \quad \ldots\ldots\ldots (107)$$

or in general

$$d_3^2 b_3 = d_{1a}^2 b_1 \mu_3 / n_o n' , \quad \ldots\ldots\ldots\ldots (108)$$

where

$$\begin{aligned} &\text{for } z_3=z_{1a}: \ \mu_3 = (N_a+N_b)\, n_a n_o / N_a \\ &\text{for } z_3=z_{1b}: \ \mu_3 = (N_a+N_b) n_a^2 n_o / N_b n_b \end{aligned} \quad \ldots\ldots\ldots (109)$$

This is not a dimensionless quantity. Either $z_3=z_{1c}=z_{1a}$ or $z_3=z_{1c}=z_{1b}$, i.e. it is assumed that pinions 1c and 3 have the same number of teeth)

The weights are

$$G_3 = d_3^2 b_3 \pi\gamma/4 = d_{1a}^2 b_1 (\mu_3/n_o n') \pi\gamma/4 \quad \ldots\ldots (110)$$

and

$$\begin{aligned} G_4 &= \varepsilon\, d_4^2 b_4 \pi\gamma/4 = \varepsilon\, d_3^2 b_3 (n'/n_o)^2 \pi\gamma/4 \\ &= \varepsilon\, d_{1a}^2 b_1 (\mu_3 n'/n_o^3) \pi\gamma/4 . \end{aligned} \quad \ldots\ldots (111)$$

making use of (65), (66), (104), (67), (105), (110) and (111), the total weight (98) is now:

$$\begin{aligned} G = [&1+(n_a/n_b)^2+\mu_1+\varepsilon(n_a/n')^2+\varepsilon\mu_1(n_c/n')^2 \\ &+2\mu_3/n_o n'+\varepsilon\mu_3 n'/n_o^3] d_{1a}^2 b_1 \pi\gamma/4 . \end{aligned} \quad \ldots (112)$$

The procedure $\partial G/\partial n' = 0$ gives

$$n'^3 - (2n_o^2/\varepsilon) n' - 2n_o^3 (n_a^2+\mu_1 n_c^2)/\mu_3 = 0 . \quad \ldots\ldots (113)$$

The value $(n_a^2+\mu_1 n_c^2)/\mu_3$ in (113) will again be designated i^*. With the aid of (103) and (109) we get for $z_{1c}=z_3=z_{1a}$:

$$i^* = (N_a n_a + N_c n_c)/(N_a+N_b) n_o \quad \ldots\ldots (114)$$

or for $z_{1c}=z_3=z_{1b}$:

$$i^* = (N_b n_b + N_c n_c)/(N_a+N_b) n_o . \quad \ldots\ldots (115)$$

If $n_a > n_b$, then $z_{1c}=z_3=z_{1a}$, or for $n_b > n_a$, $z_{1c}=z_3=z_{1b}$

Equation (113) now becomes

$$n'^3 - (2n_o^2/\varepsilon) n' - 2i^* n_o^3 = 0 , \quad \ldots\ldots\ldots (116)$$

which expression is identical with (90) and accordingly the solution is also identical with (94) or (95) as the case may be.

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[1] Hütte. Vol. IIa, 28th edition, Berlin 1954. p. 176.

[2] Gasparovic, N.: Die günstigste Aufteilung des Übersetzungsverhältnisses bei mehrstufigen Stirnradgetrieben. MuW 12(1957) 10, pp. 286/8.

Bethlehem Steel Company, Ship Building Division, Quincy, Mass.

The Navy and the Nation's first nuclear powered surface ship, the Guided Missile Cruiser *Long Beach*, underway in Massachusetts Bay. Built at Bethlehem Steel Company's Quincy Shipyard, the 721 foot long warship berthed at the Boston Naval Shipyard's South Boston Annex following the successful completion of her first sea trials. She will be drydocked at the Annex for about two weeks of routine ship's hull work after which she will return to the Quincy Yard for completion and final outfitting.

Massachusetts Bay. Built at Bethlehem Steel Company's Quincy, Mass., yard, she will be drydocked at the Naval Shipyard's South Boston Annex following the successful completion of her first sea trials. She will return to the Quincy Yard for completion and final outfitting.

TRANSACTIONS OF THE INSTITUTION OF
ENGINEERS AND SHIPBUILDERS IN SCOTLAND

SHIP DEVELOPMENTS—PAST AND FUTURE

This article was first presented as a presidential address by J. Brown on the occasion of his second election as president of The Institution of Engineers and Shipbuilders in Scotland. It is reprinted from Volume 104, Part 1, 1960-61 of the Transactions of that institution.

IN ADDRESSING YOU on this second occasion, I hope it may be of interest to review some of the historic happenings in the realm of shipbuilding and, in the light of that history, to contemplate the possibility of any phenomenal future advances.

Thirty-five years ago, when a Student of our Institution, I was fascinated listening to a paper [1] by that able naval architect, John Anderson, in which he ventured a measure of "crystal gazing." In his forecast of things to come he envisaged the arrival in the North Atlantic of the passenger liner 1,000 ft. in length. He also called attention, though without apparent conviction about the possibilities, to a suggestion [2] by A. T. Wall, that we might even see a transatlantic crossing by passenger liners of hydroplane type, or so-called "skimming dishes." We have still to see the second vision materialize, and I predict that we may never do so. On the other hand, Anderson's first prediction was not very long in being achieved, and shortly after listening to him it was my own good fortune to be engaged on the early stages of the design work for the *Queen Mary* project, which fully confirmed his foresight.

The stimulation of interest which I had experienced as a young man listening to Anderson was recalled a few months ago when one of the holders of Parliamentary office responsible for the affairs of our industry asked if I thought there was any possibility of some completely new conception in hull design which would revolutionize the present method of sea transport. He put forward, as a comparable event in aviation, the attainment of supersonic speed, although this might properly be regarded as the result of the arrival of a new means of propulsion, the jet engine. A more valid comparison might be the "break-through" from lighter-than-air dirigibles to heavier-than-air craft, or the more modern prospect of ballistic missile or spaceship travel. The question, in the context of sea transport, was not intended to refer to advances in the means of propulsion where such steps as the change from sail to steam engine, or from paddles to screws, or the introduction of the Diesel engine could be regarded as revolutionary. Doubtless the current ideas of applying nuclear power in the marine field must also be accepted as an engineering development. A more notable but as yet unattained step would be the direct nuclear provision, not merely of heat, but of electricity on an economic basis. That, however, would also be within the field of engineering progress.

Hull design appears to have shown less evidence

of distinctive and epoch-making changes than the means of propulsion. Indeed, J. M. Murray, commenting [3] on the direction and impetus given by the impact of war to disconnected advances in shipbuilding, states "there has never been with ships a new machine or method of construction, however outstanding, which on its introduction has rendered everything else obsolete." This thought-provoking statement may be offered as a challenge to our younger members who, perhaps somewhat bemused by the much publicized brilliance of new work in other technologies, feel there is no scope for their inventive genius in our more venerable industry.

Until comparatively modern times the fundamental basis for transport by sea has been a hull of the displacement type and, but for the relatively recent advent of the submarine, all surface craft. In the slow and generally gradual development from the floating log or dug-out, displacing a few hundred pounds, to the current mammoth tankers or ocean liners weighing 500,000 times as much, there have certainly been stages of progress when a remarkable step forward was taken, but Murray's thesis has an element of truth which it is not easy to refute. It is possible to point to instances where there appears to have been no progress at all. The present-day skin coracles used by fishermen in Ireland and elsewhere in this country are essentially unchanged since some of our prehistoric ancestors first evolved this particular method of building a framed hull structure. The modern small boat sailor, enthusiastic over the virtues of his catamaran compared with the normal dinghy, is repeating the discovery of the primitive man who added an outrigger to his dug-out canoe. However, this is a digression not really relevant to our thinking. We are concerned rather with the most advanced development of design or construction attained at any given period, and with the means by which it has been achieved.

The materials and constructional methods used by the boat builders of early civilizations placed restrictions on the length and attainable speeds of their craft so that exceptional advances were unlikely. The employment of the hogging truss by early Egyptians represents one device to provide the required longitudinal rigidity in the earlier raft-like boats, but the introduction of an upper deck was a more significant move towards eventual growth in size. The open Viking ships of about 500 A.D., propelled by oars and a light sail, attained lengths of 80 feet, but later Mediterranean decked galleys of about 1500 A.D. were reported to reach 130 feet in length. These later "long" ships were the warships of their day and may have been capable of speeds of 10 knots over short distances. The cargo ships were the "round" ships, having proportionately greater beam (usually as much as one-third of the length) and with an average speed of 3 to 4 knots. Throughout the Middle Ages the structural difficulties of wood construction continued to limit length of hulls, and while the change from oars to sails was taking place no great evolution of the hull could be claimed other than variations in shapes, especially of superstructures. Even in the sailing ships of the sixteenth and seventeenth centuries the length of keel did not exceed 100 feet, although the need for deck space to handle sails led to enormous overhanging fore and after castles, increasing length by 50 per cent.

By the early nineteenth century, length on waterline had crept up to nearly 200 feet for the bigger warships. Speeds for a long voyage probably averaged about 4 knots. Dr. G. S. Baker [4] reported modern tank tests carried out on a model of H.M.S. *Victory* (186 ft. by 52 ft. 4 in.) which showed that her speed at her very best was not likely to have exceeded 7 knots close hauled or 9½ knots before a fair wind.

What might be regarded as one of the greatest changes during the sailing ship era was made in the mid-nineteenth century by the Americans with their clipper ships. In these, alterations in proportions (length to beam had increased to 5.5:1) and more decisively in character of their lines, with very fine endings, made possible much higher speeds. While claims for bursts of 17 to 18 knots were made and may have been achieved on a straight course down wind, total voyage averages were normally 6 to 7 knots. British builders embraced the new ideas and *Cutty Sark* had a voyage from Australia at over 8 knots to her credit.

During the long period briefly outlined above, it is fair to say that the development of form of ships was closely controlled by practical building possibilities and in no way by any theoretical design concepts. Even the remarkable leap forward in design achieved by the Yankee clippers was not based on any accurate consideration of resistance or the like. Theoretical work there had been. About 1670 what appear to be the first systematic resistance experiments on differing forms were reported by an Englishman, Fortree, in a manuscript "Of Navarchi," while in France a professor of mathematics at Toulon published "Théorie de la Construction des Vaisseaux" in 1697. Probably the greatest landmark in the literature of the science of naval architecture was the publication in 1750 by Bouguer of his "Traité du Navire," which received well-merited attention by Prof. Robb in his Presidential Address to our Institution in 1954. Theoretical studies of the subject were also made by de Monceau, Euler and the Spaniard Santacilla. It appears, however, that these scientific or academic studies had little or no influence on the practice of contemporary ship construction. Stoot [5] attributes this to the notion that "shipbuilding was a craft with all the usual adjuncts of tradition and secrecy." He states, "Where new subjects are concerned, advances have been highly spectacular, unhampered as they have been by tra-

ditional methods," and again, "Difficulties have been few in these new industries which were not based on traditional crafts." It might be thought that in the light of these comments the absence of revolutionary changes was inevitable, at least in the nineteenth century. Stoot suggests that the gulf between the master shipwrights and the mathematicians was eventually bridged to some extent by the spread of technical education and it may be a matter of pride to us in this city that he singles out for important mention the year 1760 when an earlier John Anderson was appointed to the Chair of Natural Philosophy at Glasgow. Referring to the lectures to artisans he states, "Anderson's efforts may be regarded as the beginning of technical education and led directly to the providing of the mechanics' institutes of the nineteenth century." Stoot also pays honorable tribute to our own first President, Prof. W. J. M. Rankine, for his work in advancing the subject of naval architecture. Dr. Baker states that "the science of hydrodynamics as affecting ship form had its birth about the middle of this century" (the nineteenth) and groups as the contributors of the basic ideas Rankine, John Scott Russell and William Froude.

The early experimental studies certainly gave rise to a subsequent mass of work, both experimental and theoretical, in this field. Results have been rewarding in producing substantial savings in the power to propel normal displacement hulls and an examination of the evidence in one establishment shows over a period of 50 years, a reduction in the resistance of the average cargo ship hull of about 25 per cent, and roughly half that saving in high-speed passenger ship forms. Neither of these figures takes account of the further reductions shown by flush welded hulls in comparison with former fully-riveted ships. Reviewing marine engineering progress over the past century, Dr. T. W. F. Brown tabulates [6] evidence suggesting that reductions of over 80 per cent in equivalent fuel rates have been achieved by modern marine installations compared with the engines of the first steamships. That the naval architect has made his own, by no means negligible, contribution to the overall economy of ship propulsion is shown by the figures quoted above. Today the hull gains to be obtained show a hyperbolic trend towards a not very distant limit, even when propulsion effects are included. Moreover, it must be accepted that the improvements derived from the better co-ordination of the work of the scientists and the practitioners have been essentially gradual. It can hardly be said that any individual advances in this work have upset all previous thinking.

That the mid-nineteenth century resurgence of interest in the scientific consideration of hull design should have coincided broadly with the introduction of iron into shipbuilding almost certainly facilitated the application of the new ideas to the actual process of construction where the restrictions of wood on attainable shapes and sizes could be overcome by the use of the alternative material. It is reasonable to claim the use of iron as a major "break-through" on the hull side of shipbuilding progress. The process was very gradual and in certain sailing ship hulls a form of composite construction prevailed for a period. Nevertheless, the advance did mark a new era. Even more specifically, the *Great Eastern* (completed in 1858) should be accepted as the first example of a new concept in hull design, made possible by the change from wood to iron and by the vision of Brunel, supported by Scott Russell, in taking advantage of the possibilities in relation to the strength of this first "big-ship" hull. Such a step as the increase in length to more than twice that of the previous biggest (the *Great Britain*—also an iron ship), will certainly never occur again. The subsequent change from iron to steel and the recent partial adoption of another new material, aluminum alloy, have produced no corresponding leap forward, nor was that to be expected despite the many great advantages afforded.

Progress in hull structural design since the days of the *Great Eastern* has been continuous and, while courage and skill have been required to face each upward step in size, it might be unwise to claim any radically new idea in basic design. Have there then been no other events in the history of shipbuilding showing such an outstanding departure from previous practice as did the *Great Eastern*?

One conception which should be given a place in the catalogue of novelties in hull design, and which reached successful accomplishment towards the end of the nineteenth century, is the naval submarine. It is claimed that the first submarine, as distinct from a diving bell, was an oar-propelled vessel, presumably of wood construction, built in 1624 by Drebbel, a Dutchman. In 1775 Bushnell in America produced a submarine, driven horizontally and vertically by hand-operated screws and working on the principle of equality of weight and buoyancy with water ballast as compensation for expended weight. Fulton tried the diving plane to control longitudinal motion. Holland first employed internal-combustion engines in conjunction with electric motors and batteries for submerged propulsion and also devised adequate means of control of depth and trim. Names like Garret, Goubet and Zédé recall subsequent efforts, and by the end of the last century practicable, if still primitive, types were in service in all the principal fleets. It may be said that the submarine hull required corresponding engineering advances in the means of propulsion for its successful attainment. The recent nuclear propulsion achievements have emphasized the combined nature of the hull and machinery development. Nevertheless there is, in the conditions of submarine maneuvering, sufficient departure from

the normal ship afloat on the surface to allow us to concede that here is a design with a difference.

Although the submarine has been adopted for service exclusively in the naval field, there have been recent proposals for its commercial application. From the work of the Froudes in the experimental testing of hulls came the fuller understanding of ship resistance with its frictional and wave-making components, the latter of importance at high speeds. It is one of the attractions of sub-surface navigation that the elimination of surface wave-making resistance makes possible a substantial economy at high speeds and the skill with which American nuclear-propelled submarines have carried out many noteworthy long voyages has directed attention to the possibilities of underwater merchant ships. Many practical limitations of usefulness can be envisaged, and some of the claims for economic operation are seriously questioned. Nevertheless, the potential developments support the thought that it is not only its use in war that may justify the submarine's claim to be a new concept.

Although reference has been made to the submarine, it is not proposed to review design developments in other naval craft, except to note the modern emergence of the aircraft carrier, the use of which in the Pacific war and elsewhere shattered previous naval strategic thinking. However, it cannot be conceived to have any future contribution to make to peacetime sea transport.

The submarine seeks economy of propulsion by going below the surface. Moving in the opposite direction, attempts to break surface have resulted in the hydrofoil craft and kindred types of vessel. That this is not a wholly recent development is shown by the fact that the Scottish-born inventor, Alexander Graham Bell, working in North America and principally noted for his telephone work, also included among the many products of his fertile brain a full-size prototype of a hydrofoil craft which was tested over a period from 1911 to 1919. Photographs of passenger-carrying craft using this principle on river and short sea crossings indicate current progress but it is not easy to foresee large-scale developments in ocean-going transport of this type. It would appear to suffer from the same disabilities in relation to winter North Atlantic weather as A. T. Wall's projected "skimming dish."

The idea incorporated in the hydrofoil craft of lifting the main hull clear of the water surface to reduce water resistance has been carried a stage further in the widely-publicized new invention known to us as the hovercraft. This may justly claim to represent a radical "break-through" in the transport field. Some debate has taken place as to whether these craft should be regarded as aircraft or ships. Current development ideas appear to lean towards sea transport and in this country they have excited the attention of the Minister responsible for that department. It is of interest to note, from the paper [7] by Crewe and Eggington, that the basic idea arose from proposals by Cockerell, the inventor, to feed air under the bottom of a hard chine motor boat hull. This idea of reducing skin resistance in water by a film of air has proved a "will-of-the-wisp" to naval architects for many decades. Cockerell's novel step to the principle of the air cushion retained by side walls formed by air jets has proved a means of attaining this hitherto elusive result. Cockerell's idea of an air jet boundary was evolved after he had investigated confining the air cushion under his motor boat by longitudinal side plates, which were rejected on account of the additional water resistance they produced. This idea of solid side-wall retention of the air cushion has been revived in recent thinking, presumably on the basis that the relative resistance losses in larger craft may be minimized. Very considerable research and development work will be required in the advancement of this invention, and much interest will attend the successive stages, particularly as increase in size is attempted. Short sea ferry services may be regarded as an attainable goal, but the further target of ocean-going craft may be thought to be a more distant objective. While the invention lies in the sphere of hull design, future success will require the powerful assistance of the engineer in providing appropriate propulsion and air-cushion arrangements to supplement the naval architect's efforts in relation to hull forms, strength, safety equipment and other associated problems.

It is possible to cite many examples of original thinking in the introduction of special types of ships. Most notable among these in its impact on the nature of the world's fleets was the advent, about the 1880's, of the bulk oil carrier. After the initial step from storage in internal tanks to the bold idea of using the hull itself as the storage tank, evolution has been rapid. The subequent growth in size to the super-tanker of today and the volume of the tonnage in tanker fleets, confirm the importance of this particular venture.

Another innovation with ocean-wide repercussions was the experiment of carrying frozen cargo in ships' hulls. While this was obviously a co-operative effort on the part of the chemical scientist and the practical shipbuilder, or in modern days of the refrigerating engineer and the insulating specialist, the first adaptation of hold spaces on cargo ships of the late '70's and early '80's was as productive of a new era in sea transport as the early tanker building.

A novel design may be sponsored by one forward-looking firm, and the Doxford trunk deck ships of the 1890 period are in this category. The design had as its essential purpose the provision of a strong hull girder with clear holds and a dry weather deck. The idea had a passing phase of popularity which has not, however, endured.

Specialized types receiving some favorable publicity in recent years are the so-called "roll-on-roll-off-ships" and, for wider use, "container-ship" designs, but these are not likely to take a major place in the main overseas transport of cargo. More significance should perhaps attach to the growing popularity with shipowners of the large bulk cargo carrier, which appears to have been derived from earlier ore carrier types and to have promise of being as appropriate for a large part of the world's cargo trade as the oil tanker became in its particular field.

In this examination of new inventions or ideas which might earn the distinction of being "different," attention has been concentrated on the overall hull design. There have been certain features of detailed structure or equipment which could doubtless be regarded as in the category of major improvements.

The outstanding success of the introduction of longitudinal framing (due essentially to the efforts of Isherwood) is still bearing fruit in application beyond its original use in tanker structures, while the attainment of the giant tankers of present-day dimensions would hardly have been possible without the benefit of this structural feature.

An even greater impact on hull structures has resulted from the application of electric welding in place of riveting. This modern revolution, with its consequential effects on constructional methods and the increased freedom in overall and detailed design which it permits, may rank as equal in importance to any other single step forward, perhaps even to the introduction of iron.

Streamlined rudders, tunnel sterns, Kort nozzles, side-thrust propellers, might all come into the classification of advances in their own specialized applications. Bulbous bows as a means of reducing power would also stake a claim, even if they were incorporated for other reasons in much earlier times. The fin-stabilizer, promoted by its Scottish sponsors, Denny-Brown, to the position of an essential in all major passenger-carrying vessels would certainly feature in such a list.

Some other inventions have proved less enduring. Mr. S. W. Bolwell, a Member of our Council, recently entertained a joint meeting of the Greenock Association of Engineers and Shipbuilders and this Institution with a paper on "Freak Ships," in which descriptions were given of some unusual designs of this type. Articulated hulls to ride the waves, double hulls to give stability, and a hull incorporating a saloon mounted on gimbals were typical examples of the less successful schemes which were mostly imaginative projects without any scientific backing.

In the foregoing somewhat discursive study an attempt has been made to evaluate some of the more remarkable developments in the field of shipbuilding, as distinct from that of marine engineering, and to set them against the background of the contemporary theoretical knowledge. It shows that the link between the constructors and the scientists was for long virtually non-existent, but for at least a century past they have realized their interdependence and, in spite of some ill-informed comments to the contrary, there cannot be many technologies today in which there exists such a continuous and rewarding application of theory to practice.

Many men of distinction, both in the academic and in the practical field, have gained well-merited renown for their individual contributions to the art and science of shipbuilding although it may be thought that few could claim to have sponsored such an unprecedented advance as that of Brunel with his monumental *Great Eastern*. While it is true that some, like Cockerell, have had their flashes of inspiration, progress to perfection tends to be achieved more usually by vigorous leadership controlling patient and persistent team work in the application of all the relevant knowledge. Institutions such as our own have done much to promulgate that knowledge; it is their major purpose. Our British research associations are second to none in their efforts to discover and extend the scientific data for future advances. The industry in this country has never lacked, nor does it lack today, the requisite technical skill and active direction to ensure continued progress.

Parkinson in his recent book [8] discussing the factors that influence the competitive position of our industry, states "Pride of place must, even in this era of highly-organised production, be given to inventive genius." Such words from an economist should surely offer to shipbuilders and engineers the greatest measure of encouragement and stimulus to their efforts for future progress.

What then is the answer to my Parliamentary friend's query on the possibility of some entirely novel innovation in ocean transport? Will Murray's view of a gradual evolution prove correct, or will some spectacular development, perhaps in the mind of one of our younger members present tonight, prove this view untenable? My own opinion is that nothing in immediate prospect or likely to appear will wholly displace the ships we are now capable of producing and which will constitute the most important means of transport over the globe. Continued improvement is to be expected and should be ceaselessly sought and extended to the limit of our capabilities. The application of our inventive genius to this task affords the most likely source of any further major advance and offers a perpetual challenge to us.

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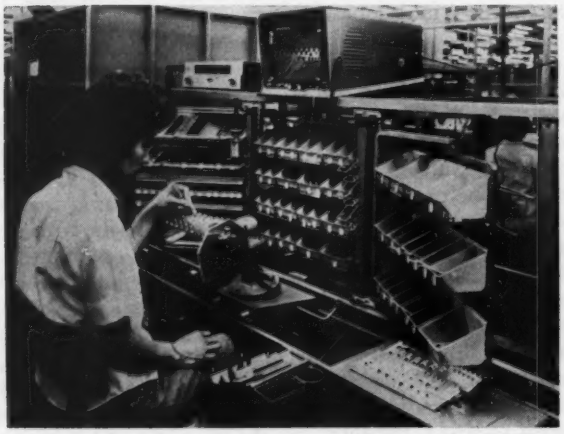
AUDIO-VISUAL AID SPEEDS PRODUCTION



Engineers at General Dynamics/Pomona have brought audio-visual aids into a missile plant and speeded the production of complex electronic assemblies by 40 per cent. Maximum efficiency and speed in assembling electronic equipment are always of critical interest particularly in the electronic guidance elements of the Advanced Terrier and Tartar missiles planners. The Pomona plant is organized to shift into full production of these or of whatever may be required within hours of a national emergency.

The audi-visual devices used in the tests combine colored slides with a taped sound track. The operator observes the slides on an eye-level screen, and at the same time listens through ear phones to a detailed description of the assembly steps to be followed. The slides and tape are controlled by the worker—preventing any arbitrary speedup. The worker is allowed to complete each operation at his own working pace, then move on to the next assembly sequence.

First step in preparing for the tests was to convert from a line-flow, multiple-worker assembly procedure. Next a series of production and assembly photographs were taken and mounted as slides. These were reviewed with the workers who would be using them. Finally, a taped series of instructions was recorded and coordinated with the slide photographs.

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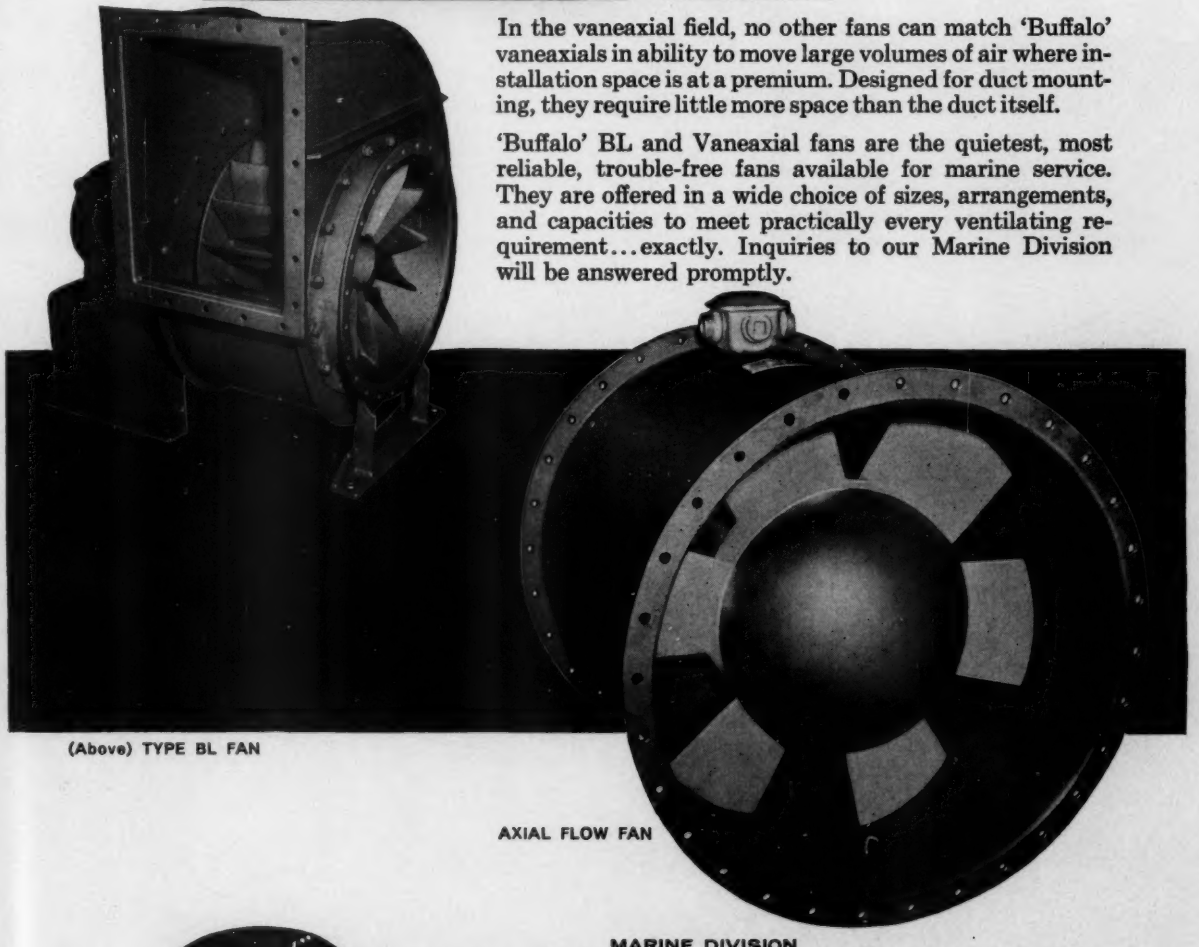

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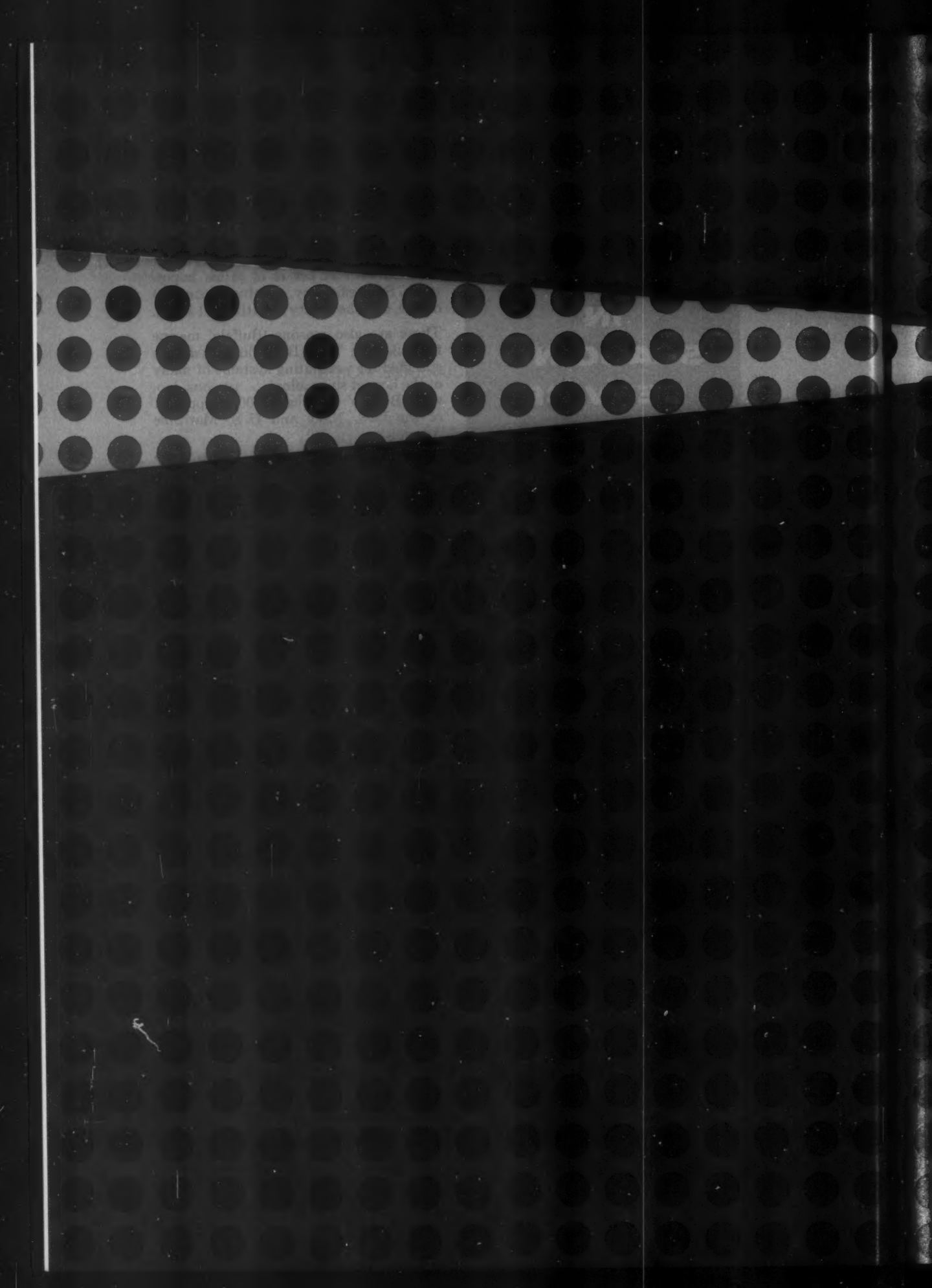

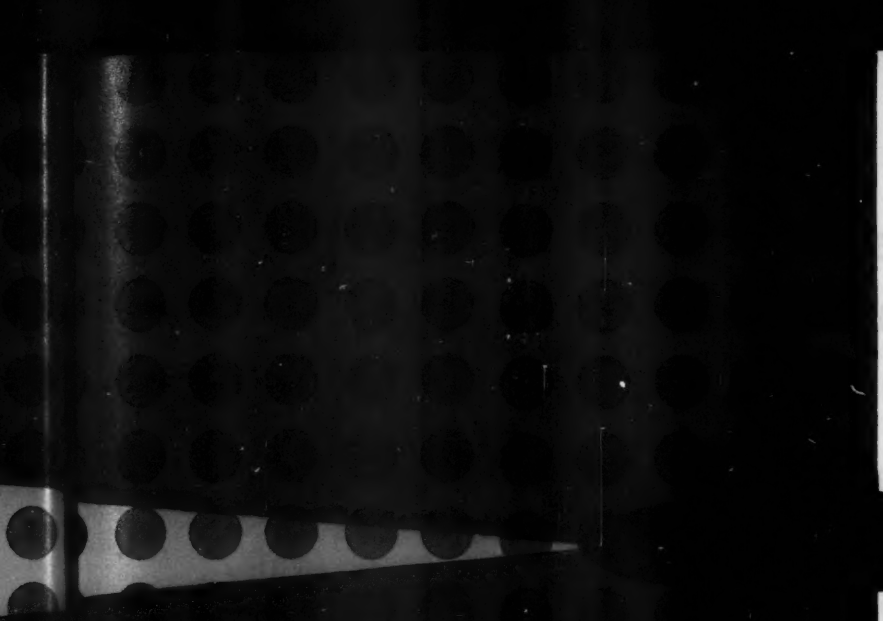

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ROGER S. PORTER AND JULIAN F. JOHNSON

FLOW PROPERTIES OF MODERN LUBRICATING OILS

THE AUTHORS

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VISCOSITY IS THE MOST important property of a lubricating oil. It determines the load-carrying capacity and power losses in bearings. The lubrication requirements of modern machinery have steadily grown more severe. To meet these requirements, new lubricants have been developed. Many of these contain high molecular weight polymers; and, as a consequence, their flow properties become complex. The purpose of this article is to illustrate some of the types of behavior that can occur in these multicomponent systems with particular emphasis on non-Newtonian flow properties.

EXPERIMENTAL

Newton, among his many accomplishments, defined the resistance of a liquid to flow as

$$F = \eta \, dV/dx \quad \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

where F is the shear stress, dV/dx is the shear rate, i.e., the rate of change in velocity with the distance from the wall; and η is the viscosity coefficient, now shortened to "viscosity," with units in poises. Fluids that follow this relationship are called Newtonian. Measurements in many types of instruments give results that are in terms of viscosity divided by density. This is called the kinematic vis-

cosity, and the units are called stokes.

There are numerous ways of determining viscosity; and an extensive description of these methods is outside the scope of this article. However, most of these methods fall into two principal classes. Capillary instruments are based on Poiseuille's law for flow in capillary tubing. This equation,

$$Q=\frac{\Delta P \pi R^4 T}{8\eta L} \quad \ldots\ldots\ldots\ldots\ldots\ldots (2)$$

where Q is the volume of liquid flowing through the capillary in time, T, ΔP is the pressure drop across the capillary of radius, R, and length, L. In principle, all capillary instruments measure the time necessary for a given volume of liquid to flow through a capillary. The measured results are in terms of kinematic viscosities. The shear rate in a capillary varies continuously from zero at the center of the capillary to a maximum value at the wall following a parabolic distribution for Newtonian liquids. At the entrance and exit zones of the capillary, the flow is not uniform; and end corrections become important. These end corrections become increasingly important as the rate of shear increases.

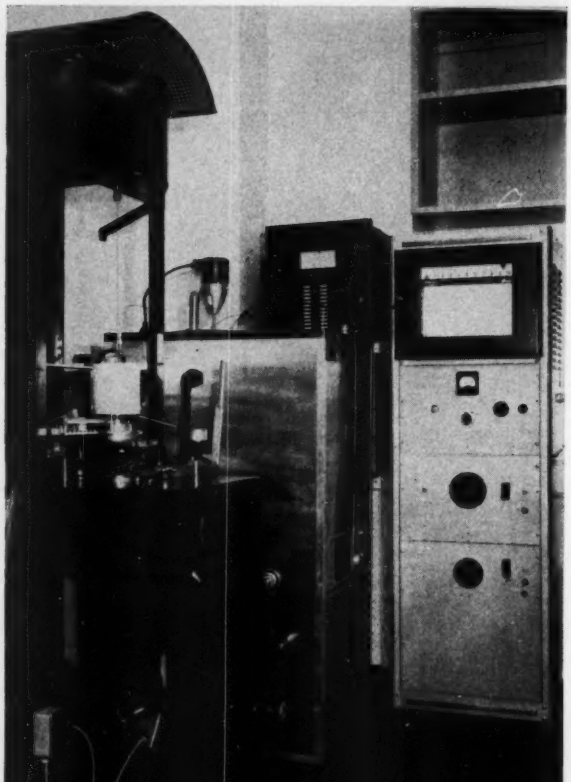
The second principal class of viscometers is the concentric cylinder type. In these viscometers, a film of liquid is contained between two concentric cylinders, one of which is rotated at a known speed, which defines the shear rate. The resulting torque on the other fixed cylinder gives a measure of absolute viscosity. In general, these instruments are more complex than the capillary type; but they have the advantage of approximately constant shear rate if the annular gap is small compared to the radius of the inner cylinder. Thus, they are particularly well suited to measuring viscosity as a function of shear for non-Newtonian fluids. Recently, concentric cylinder viscometers have reached a high state of refinement. A photograph of the instrument on which all the results in this article were obtained is shown as Figure 1. It is based on a design by Barber, et al, and has been fully described elsewhere [1, 2].

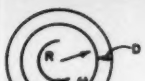
Viscosity changes sharply with temperature. For many liquids, this relationship may be expressed as

$$\eta=A^{e^{B/T}} \quad \ldots\ldots\ldots\ldots\ldots\ldots (3)$$

where T is the absolute temperature and A and B are constants. Such equations must be applied with care, however; and it is better to measure the viscosity at the temperature of interest rather than to extrapolate it from measurements at other temperatures. The viscosity behavior changes sharply if a second phase is present. For example, if the temperature is low enough so that a solid forms, the flow behavior of the systems is very complicated as it often depends on the thermal history of the sample. This article will be confined to single-phase systems.

It is desirable for a lubricating oil to show a small dependence of viscosity on temperature. It is difficult to describe the viscosity-temperature relationship without giving the actual experimentally determined curve. In order to have a relative measure of this temperature dependence, the "viscosity index" has been devised. The viscosity index is an empirical number indicating the effect of change of temperature on the viscosity of an oil. A low viscosity index indicates a relatively large change of viscosity with temperature. In order to improve the viscosity-temperature characteristics, high molecular weight polymers are often added to lubricating oils. Table I shows the marked increase in viscosity index that can be obtained in this manner.





SHEAR RATE $=\frac{wxR}{D}$

Figure 1. High Shear Viscometer.

Additives of this type are responsible for the modern multi-grade lubricating oils such as the 10W-30, 20W-40, and 5W-30. These oils meet the viscosity specifications for the several grades because they are below the maximum set at 100°F and above the minimum grade requirements at

TABLE I—*Viscosity Index as a Function of Polymer Added and Shear Rate*

Polyisobutene Molecular Weight	Per Cent Polymer in Cetane	Regular Capillary Viscosity Index	Viscosity Index at Constant Shear Rates Reciprocal Seconds		
			10^4	8×10^4	8×10^5
	0	127	127	127	127
4.0×10^6	10	174	—	180	196
6.2×10^6	10	174	175	180	193
2.2×10^6	10	<125*	130	—	—

* Non-Newtonian in capillary viscometer at shear rates below 100 sec^{-1}.

210°F. Thus, one grade of lubricant can meet the requirements that formerly necessitated inventorying several grades. However, when such additives are used, the oils often exhibit non-Newtonian flow characteristics. They will be examined in detail in the following sections. It can be seen from Table I that the viscosity index depends on the shear rate at which the viscosities are measured and usually is higher at high shear rates. This is an additional reason for the superior performance of polymer-thickened oils.

NON-NEWTONIAN FLOW PROPERTIES

There are two general non-Newtonian flow properties associated with polymer solutions, the reversible and irreversible effects. The reverisible effect consists of a decrease in viscosity with increasing shear. When the shear stress is removed, the viscosity reverts to its original low shear value. This effect is caused by the disentanglement of polymer chains and aligning of polymer molecules due to the shear stress. This causes an effective reduction in the resistance to flow. The irreversible effect is a permanent decrease in viscosity after an oil has been subjected to a high shear stress. This decrease is due to the rupture of polymer bonds, thus lowering the molecular weight of the polymer.

A. *Reversible Viscosity Effects*

Figure 2 gives an example of the lowering of viscosity with increasing shear stress for a polymer-thickened multigrade oil. For comparison, the lack of effect of shear on the Newtonian base oil without the polymer is also shown. From Figure 2, it can be seen that the change in viscosity with increasing shear can be very large and occurs at shear rates at or below those occurring in actual machine elements of interest. Eventually, a shear rate is

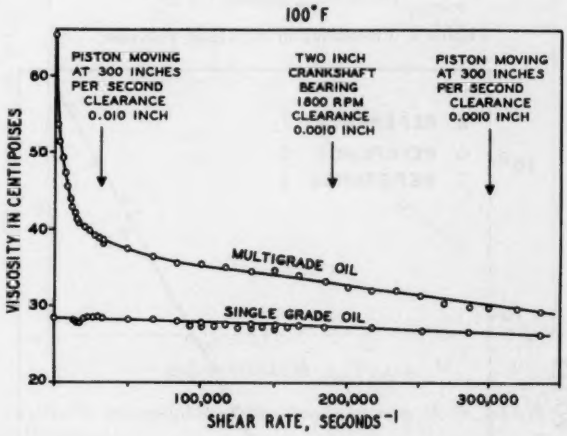


Figure 2. Data from High Shear Viscometer.

reached where a permanent decrease in viscosity will occur; but below this shear, the effect is entirely reversible. Figure 3 shows the decrease in viscosity of bulk polymers, that is with no solvent added, as a function of shear. It can be seen that below some molecular weight, the change is negligible; whereas, at higher molecular weights, the change is quite large. In the example, for instance, the viscosity decreases from about 108 poises to below 10 poises. It has been shown, for some polymer systems, that plots of the log of the viscosity of polymers measured at low shear rates versus their molecular weight exhibit a discontinuity at some molecular weight that corresponds to the point where chain entanglement of the polymers becomes important [3, 4]. Such a plot for polyethylenes is shown in Figure 4. For the polyethylene system, this entanglement molecular weight corresponds ap-

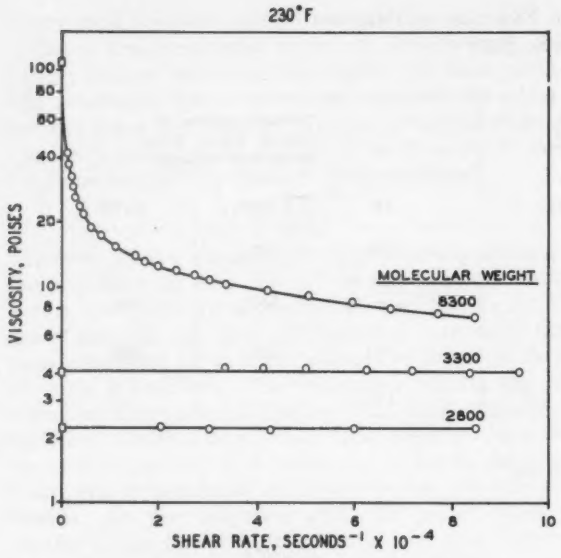


Figure 3. Viscosities of Ethylene Polymer.

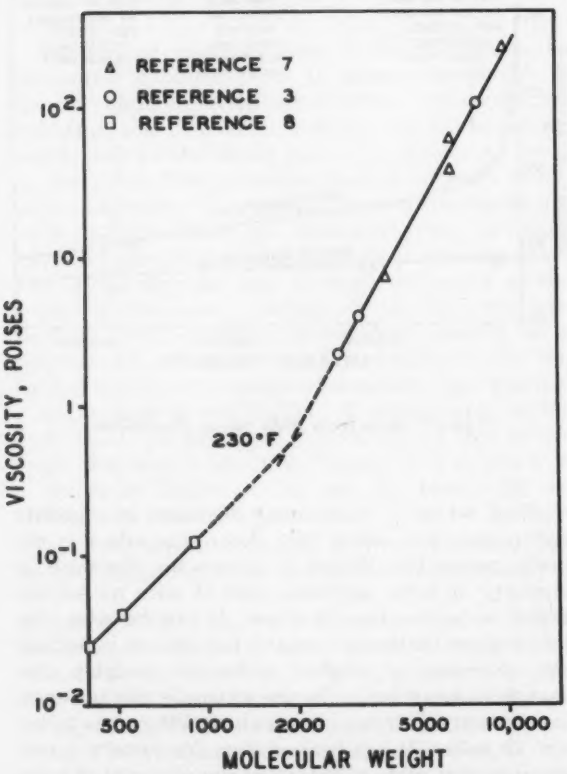


Figure 4. Low Shear Viscosities of Polyethylenes and Normal Alkanes.

proximately to the point where the reversible effects are observed. Thus, low shear viscosities can be used to determine the molecular weight of polymer above which non-Newtonian effects will become important.

These reversible viscosity decreases are often of importance. In an internal combustion engine, for example, the oil layer between the piston and the cylinder is being subjected to a relatively high shear. The viscosity and, therefore, the power loss due to friction will decrease. The loss of oil, however, on the stationary nonsheared cylinder walls can be correlated with the viscosity. The higher viscosity at low or zero shear, therefore, results in a much lower oil consumption. In effect, one can have one's cake and eat it also. The lower viscosity at high shear reduces friction losses without increasing the consumption as would be the case if one used a Newtonian oil with the low viscosity of the compounded oil at high shear. Figure 5 shows other illustrations of change of viscosity with shear for a polyisobutene at various temperatures [5].

A possible way of estimating the maximum decrease in viscosity with increase in shear from low shear data is shown in Figure 6. This is a plot of capillary viscosities at a given concentration of polymer for various molecular weights of the polymer. The limiting slope at the lower molecular weights is extrapolated and assumed to be the lowest value of viscosity that will be obtained from increasing shear rate. This seems to be valid for the limited number of systems tested to date.

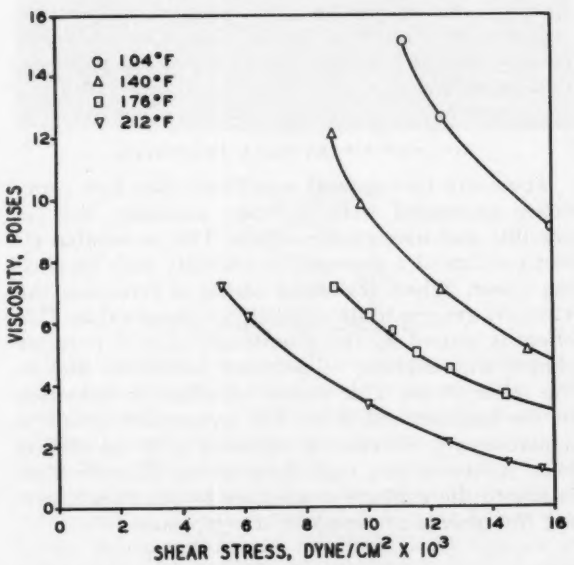


Figure 5. Viscosity of 10% Polyisobutene M. W. 2.2×10⁴, In Cetane.

B. *Irreversible Viscosity Effects*

Under conditions of laminar flow, it is possible to produce irreversible decreases in viscosity at high shear by the rupture of polymer bonds with a subsequent decrease in the average molecular weight. This results in a change in the reversible viscosity shear properties. Generally, it also causes an in-

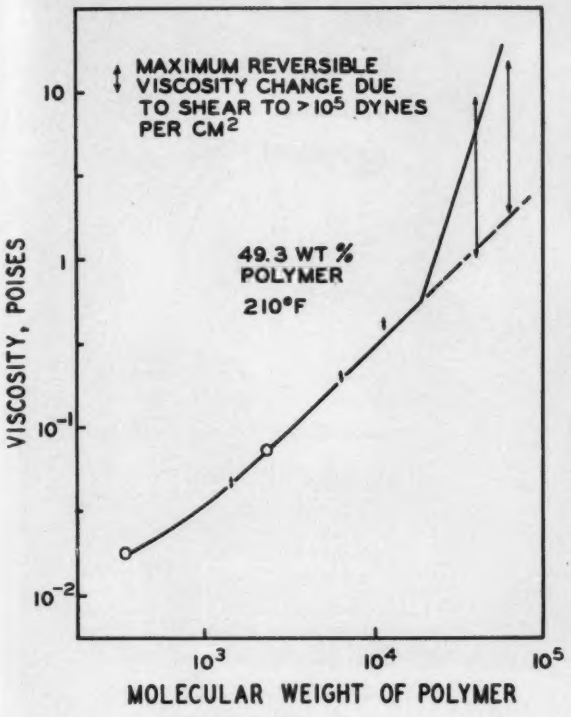


Figure 6.

crease in oil consumption. Figure 7 shows an example of laminar flow degradation of polyisobutene. There is no irreversible change until a certain minimum shear rate or shear stress is reached. Then, for any given shear stress, the polymer degrades until an equilibrium molecular weight is reached. As the shear increases, this equilibrium molecular weight decreases. For polyisobutene, this effect seems to depend only on the shear stress, not on the temperature. Figure 8 shows the equilibrium molecular weight attained at various shear stresses at three temperatures. The results are superimposable.

Again, the concentric cylinder viscometer offers an excellent experimental technique for studying laminar flow degradation. The shear rates are well defined, in contrast to the capillary-type viscometer where the shear rate is not uniform and degradation is dependent on the capillary geometry.

CONCLUSIONS

Polymer-thickened oils offer many advantages. They reduce cold starting problems and decrease power losses due to friction. Design engineers, however, must consider the effect of shear rate on the actual observed viscosity in the machine element and the possibility of irreversible effects due to mechanical degradation of the polymers.

While not extensively developed, there is some indication that these effects can be predicted from low-shear capillary viscosity measurements on polymer solutions if polymers of various molecular

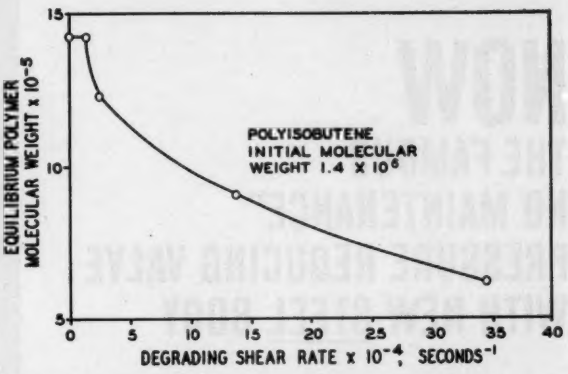


Figure 7. Degradation of 10% Solutions of Polyisobutene in Cetane at 40°C.

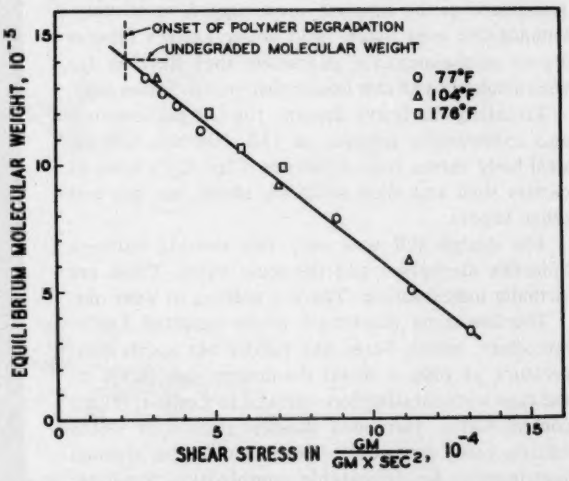


Figure 8. Degradation of 105 Polyisobutene, M. W. 1.4×10⁶, in Cetane.

weights are available. This greatly reduces the amount of experimental work required to define the flow properties.

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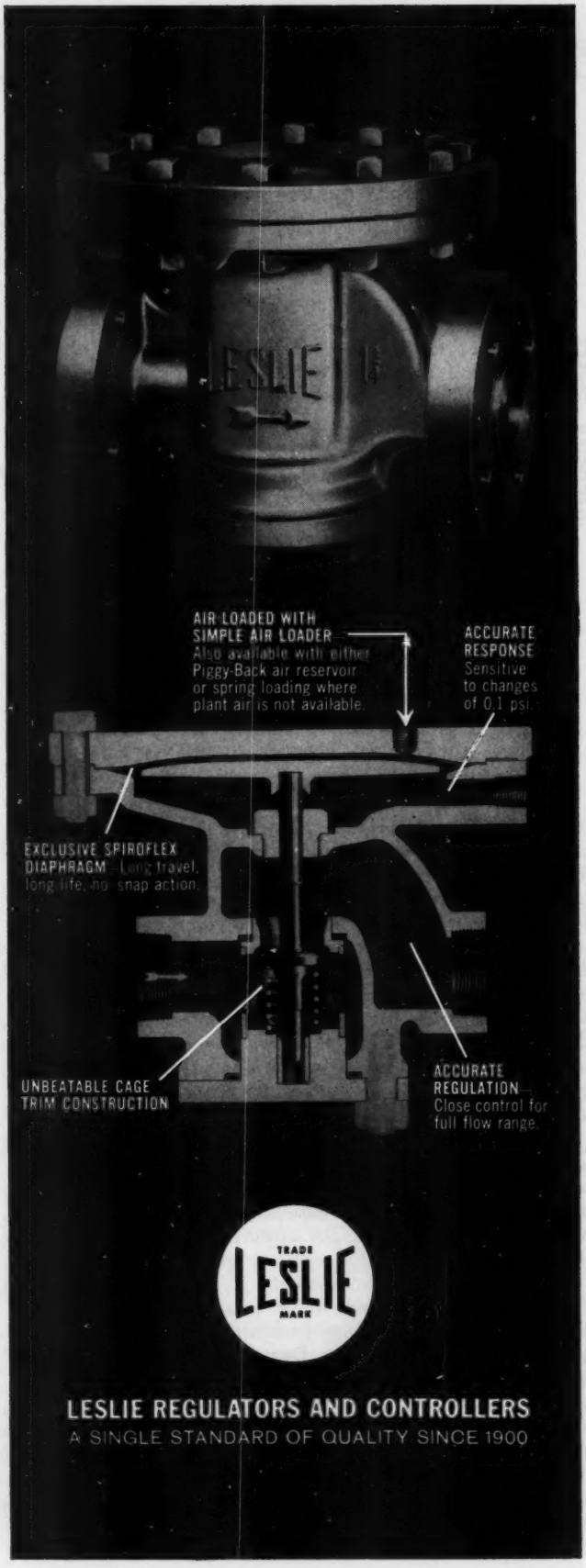
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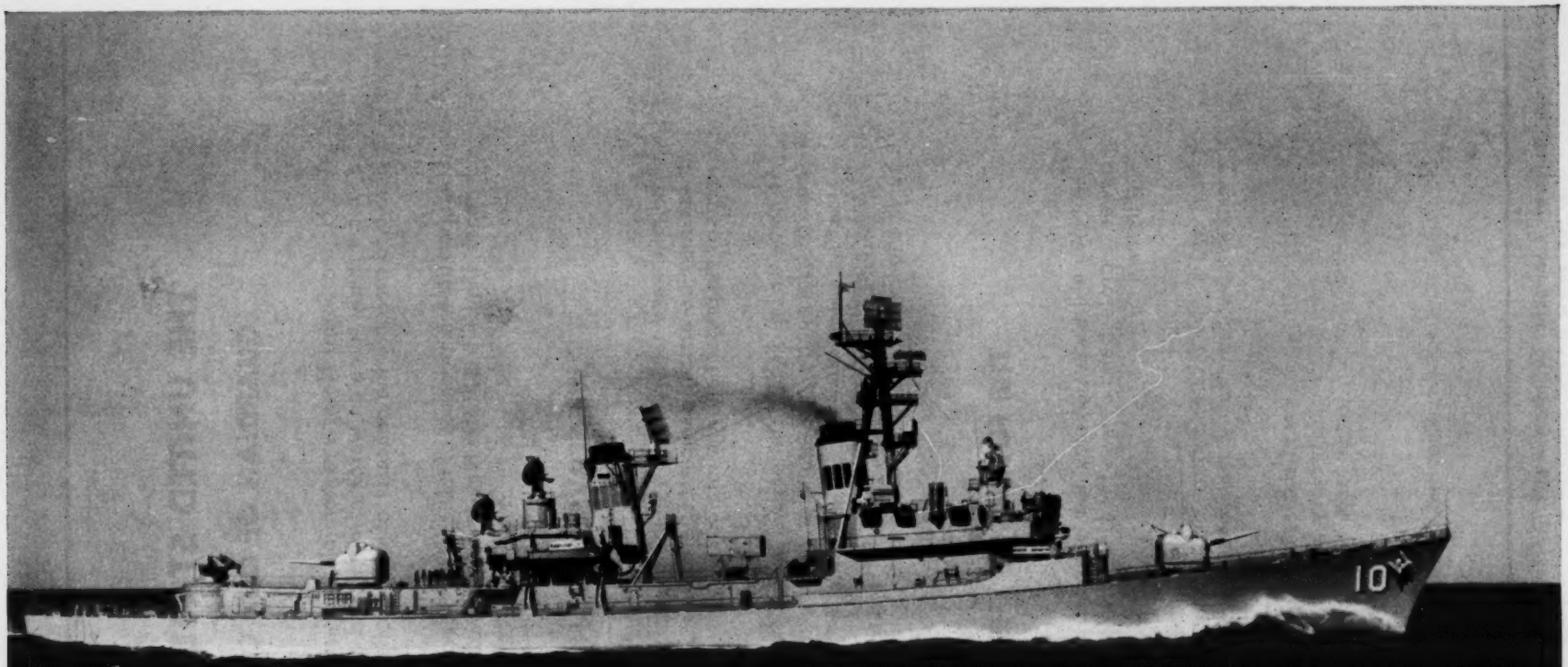

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RECENT RESEARCH ON FATIGUE IN METALS

ACKNOWLEDGMENT

This review of progress in metal fatigue over the last five years, by P. G. Forrest, Ph.D., appeared in the March 1961 issue of "The Chartered Mechanical Engineer," from which it is reprinted.

FATIGUE IS A TYPE of failure that can occur in almost all metals and non-metallic materials as a result of the application of varying or repeated loads. Its importance in mechanical engineering can scarcely be overrated, for it accounts for most failures in service. This is because a metal will fail by fatigue from the application of an alternating stress that is considerably lower than the static tensile strength and often lower than the yield stress. Consequently, there is often little deformation in the region of a fatigue fracture and this is one of its most striking characteristics.

A further consequence is that fatigue failures almost always propagate from some surface geometrical discontinuity, because any change in cross-section of a loaded part, such as a hole, a fillet, or even a keyway, causes a local increase in stress or stress concentration. A photograph of a typical fatigue fracture in a crankshaft, is shown in Figure 1.

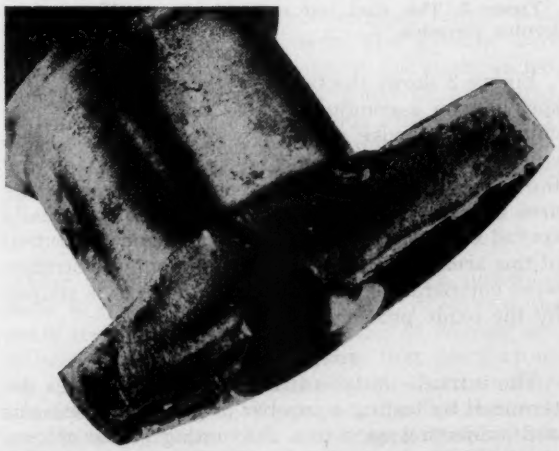

Figure 1. Fatigue failure in a crankshaft, due to stress concentration between journal and web.

Failure has resulted from the stress concentration at the fillet between the journal and the web.

The static strength of a ductile material is little affected by such changes in section, because the stress concentrations are relieved by plastic deformation, but the resistance to fatigue failure is considerably reduced because the range of stress is almost unaffected by deformation. Because of this sensitivity to stress raisers, great care is needed in the design of parts required to withstand fluctuating stresses, in order to reduce the magnitude of all stress concentrations sufficiently to prevent fatigue failure. In general, the severity of a stress concentration can be reduced by reducing the curvature, which means, in practice, avoiding sharp corners and providing generous fillet radii.

While allowance for stress concentration is of paramount importance, many other factors must also be considered if fatigue failure is to be avoided. The fatigue strength of a part is dependent not only on its shape but also in its size and, in addition, on the mode of stressing and the ratio of static to fluctuating loads. Furthermore, metals are particularly susceptible to fatigue failure in a corrosive environment and their resistance is reduced much more drastically by the combined effects of corrosion and fluctuating stress than by either factor acting singly. Fretting corrosion is an even more prevalent source of fatigue failure. This is a process of abrasion resulting from small repeated relative movements between clamped parts. It occurs, for example, in riveted and bolted joints that are subjected to fluctuating loads and can reduce their fatigue strengths to very low values indeed.

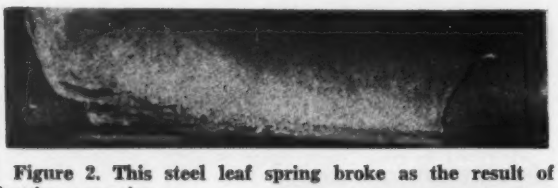

Figure 2. This steel leaf spring broke as the result of fretting corrosion.

Figure 2 shows the fracture surface of a steel leaf spring from a combined stress fatigue testing machine which broke where it was clamped, as a result of fretting corrosion. The smooth zone covering the greater portion of the fracture surface is the area across which the fatigue cracks have gradually spread before final fracture occurred and the extent of this area is an indication that the applied stresses were comparatively low. Part of this area is stained by the oxide produced during fretting.

FATIGUE TESTS

The intrinsic fatigue strength of a material is determined by testing a number of polished specimens and subjecting each to a fluctuating stress of constant amplitude until it breaks. By applying a different stress to each specimen, a relation is obtained between the applied stress and the number of cycles to failure. The results are presented on a stress-endurance ($S-N$) curve and Figure 3 shows $S-N$ curves for an alloy steel and a carbon steel. The fatigue strength is defined as the alternating stress that the material will withstand for a given endurance. It will be seen that no failures occurred in the carbon steel after 3 million cycles, even though 5 tests were continued for 100 million cycles. The $S-N$ diagram for this material can therefore be drawn as a curve through the points representing fractured specimens and a horizontal line above those representing unbroken specimens. The stress at which the curve becomes horizontal is known as the fatigue limit. A further feature of fatigue behavior, illustrated in Figure 3, is the considerable scatter in the alloy steel results. There is more scatter in fatigue strength than in other mechanical properties and it is particularly marked in high strength alloys [1] [2].

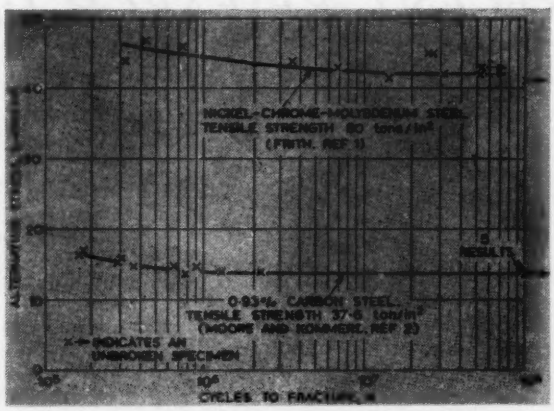

Figure 3. S—N curves for steels: a relationship between applied stress and the number of cycles to failure.

Fatigue tests can be made with superimposed static or mean stresses, at high or low temperatures, in a corrosive environment or on specimens containing stress concentrations in order to provide a closer approach to service conditions. Even so, the results of such tests are often inadequate for predicting the fatigue strength of engineering parts and for this purpose it may be necessary to carry out tests on actual joints, or even on full-scale structures.

An International Conference on Fatigue of Metals was held at the Institution in 1956 [3] and this provides a useful starting point for reviewing the recent progress in research on the subject. The development up to that time was reviewed by Gough in his introductory address to the conference. No less than 81 papers were presented and this is indicative of the wide interest in fatigue. These were divided into the following subjects for the purpose of discussion: stress distribution, temperature, fre-

quency and environment, metallurgical aspects, basic aspects and engineering and industrial significance.

STRESS DISTRIBUTION

The results shown in Figure 3 were obtained with the stress alternating between equal values of tension and compression, so that the mean stress was zero. It is found that if a mean tensile stress is superimposed, so that the stress fluctuates wholly in tension or between unequal values of tension and compression, then the range of stress that a material can withstand for a given endurance is reduced. The addition of a mean compressive stress, on the other hand, increases the fatigue strength. This is illustrated in Figure 4, using results presented at the Fatigue Conference [4] [5]. Why metals behave in this way has not been clearly established, but it may be that a compressive stress inhibits crack propagation, while a tensile stress accelerates it. Whatever the explanation, it is of considerable practical importance because it means that fatigue strength is strongly influenced by residual stress.

Fatigue failures almost always propagate from the surface so that the resistance of a part to fatigue failure may be reduced if there is a residual tensile stress at the surface, and may be increased by a residual compressive stress at the surface. Surface residual compressive stresses can be induced by a number of surface treatments, which include shot peening and surface rolling, case-hardening and flame- and induction-hardening and this accounts for some of the improvements in fatigue strengths obtained by these treatments [6].

In low strength materials the beneficial effect arises mainly from the intrinsic strengthening of the surface layers, for the residual stresses tend to disappear as a result of yielding during subsequent stressing, but in high strength metals the effect of residual stress may be predominant. A convincing demonstration of this has been provided recently by Mattson and Roberts [7] by means of strain peening experiments on a spring steel, treated to a Rockwell C hardness of 48.

Strain peening is a modified form of shot peening in which the surface of the specimen is subjected to a tensile strain by bending, while it is bombarded with iron or steel shot. This induces a higher residual compressive stress in the surface than shot peening alone, with a consequent improvement in fatigue strength. The adverse effects of some plating processes—particularly nickel and chromium—on fatigue strength, can be attributed to tensile residual stresses at the surface and improvements are obtained if these stresses are reduced [8].

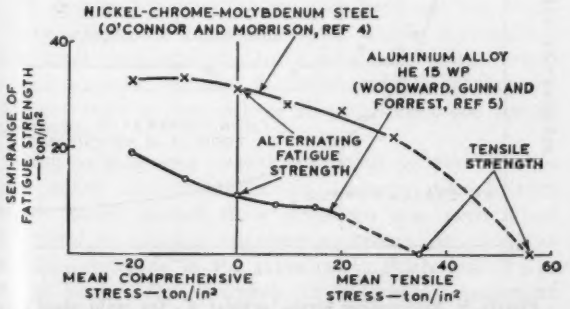

Figure 4. Effect of static stress: a mean compressive stress actually improves the fatigue strength.

FATIGUE OF THICK CYLINDERS

It might be expected, from the influence of mean stress, that fatigue failure is dependent on the maximum value of the tensile stress but, for ductile metals, it depends predominantly on the shear stress. This is evident from a comparison of fatigue strengths in bending and torsion; in addition, metallographic observations show that fatigue cracks originate within slip bands, which are produced in the metal by a shearing action. Further evidence has recently been provided by Morrison, Crossland and Parry [9] by means of tests on thick cylinders subjected to repeated internal pressure. The stress system in a cylinder can be regarded as a uniform triaxial tension superimposed on a simple shear stress which has maximum value at the bore. The ratio of the triaxial tension to the shear changes with the ratio of the external to the internal diameter of the cylinder. Morrison, Crossland and Parry tested cylinders of 7 different materials, all with a 1 inch bore, but with wall thicknesses ranging from 0.1 to 1 inch, and the results showed quite clearly that the fatigue strength was dependent almost entirely on the range of maximum shear stress and not significantly on the magnitude of the triaxial tension.

The main practical significance of the results was that the resistance of the cylinders to fluctuating internal pressure was considerably less than expected from the results of conventional fatigue tests in torsion on small specimens. The authors were able to show, at least for one of the steels tested, that some of the discrepancy could be attributed, in some way, to the deleterious effect of the oil used to obtain the internal pressure, for a considerable improvement in fatigue strength was achieved by protecting the bore with a rubber coating.

VARYING AMPLITUDE

One of the problems encountered in designing parts to resist fatigue failure is that, in service, the parts are usually subjected to loads of varying amplitude. If it is possible to ensure that the fluctuating stresses never exceed the fatigue limit or long-life fatigue strength, design may be based simply on fatigue strength. For some applications, however, this is not possible and the stresses will sometimes exceed the fatigue limit. On this basis, fatigue

failure must be expected if the part is used for a sufficiently long time and the design must therefore be based on a finite life.

For this purpose, it would be useful to be able to predict the fatigue life under stresses of varying amplitude from the results of conventional fatigue tests at constant stress amplitude. The linear damage law provides the simplest basis for doing this. The assumption is made that the application of n_1 cycle at a stress range S_1 for which the number of cycles to failure is N_1 causes an amount of "fatigue damage" n_1N_1 and that failure will occur when $\Sigma n_1/N_1=1$.

This relation is useful for making an approximate estimate of the fatigue life but does not provide a reliable means of determining the life under all varying stress conditions. There are a number of reasons for this; firstly, the fatigue strength of some metals, notably steel, can be considerably increased by repeated stressing below the fatigue limit; secondly, the occurrence of a few heavy overloads on a component are likely to introduce residual stresses at stress raisers and, thirdly, fatigue cracks produced by high stresses can be propagated by stresses well below the original fatigue limit. Some attempts have been made to derive non-linear laws [10], but in view of the influence of the above factors, no law of general application can be expected. It is now generally accepted that when a part is designed for a limited fatigue life, it is wise to confirm the estimate by subjecting full-scale parts to program tests, that is, to a sequence of stress amplitudes, applying a number of cycles at each stress in proportion to the distribution occurring in service [11].

CRACK PROPAGATION

Until recently, there had been little reliable data on the rate of propagation of fatigue cracks, partly because of the difficulty of interpreting measurements made on small specimens. An investigation in progress at the National Engineering Laboratory is, therefore, of particular interest, for it shows that, in the early stages, the rates of crack propagation for a wide range of materials can all be expressed by one basic equation [12] [13]. The tests are made on 10-inch-wide sheets, about 0.1 in. thick, containing a small central slit. These sheets are subjected to fluctuating tensile loads. It has been found that the crack growth may be discontinuous, particularly at low ranges of stress, but when the growth is continuous, the growth rate for crack lengths up to one-eighth of the sheet width can be represented by the following equation for all the materials tested:

$$\frac{dl}{dN} = \frac{lS^3}{N_s}, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

where l is half the length of the crack including the initial slit, S is the nominal alternating stress on the cross-section, N is the number of cycles and N_s is a material constant which may depend on the mean stress. The rates of crack growth for the materials tested, relative to the rate for mild steel, are summarized in Table 1 [13]. The susceptibility to crack propagation does not appear to be related to other mechanical properties. As a crack proceeds, a stage is reached at which the relatively slow rate of propagation described by equation (1) suddenly changes to a much higher rate which quickly leads to fracture.

Frost has also made a study of the conditions governing the non-propagation of fatigue cracks [14] [15]. Cracks are sometimes found to develop at the root of very sharp notches, reach a certain size, but do not propagate further. This behavior can be illustrated by Frost's results on notched specimens of mild steel, which are summarized in Figure 5. All the specimens contained notches 0.05 in. deep, but radii at the root of the notch, varying from 0.09 in. to as little as possible were used in order to vary the severity of stress concentration. A measure of this severity is given by the stress concentration factor, K_t, which is defined as the ratio of maximum local stress in the region of a discontinuity or notch, to the nominal stress evaluated by simple theory.

Frost found that the nominal stress required to start a fatigue crack at the root of a notch was in good agreement with the theoretical value, given by the unnotched fatigue limit, which was 16¾ tons/in², divided by K_t and this is represented by the curve in Figure 5. However, if the nominal stress (based on the net area) was less than 5¾

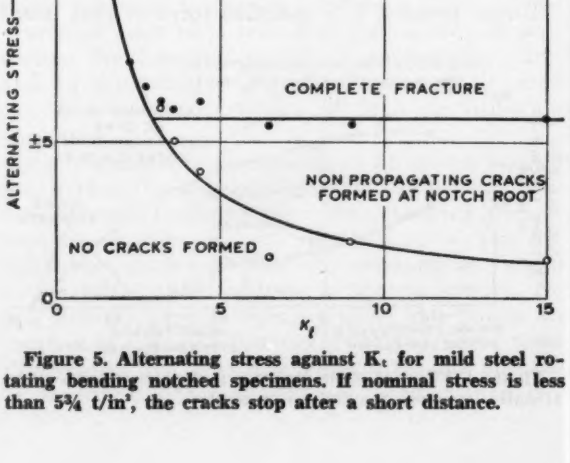

Figure 5. Alternating stress against K_t for mild steel rotating bending notched specimens. If nominal stress is less than 5¾ t/in², the cracks stop after a short distance.

TABLE I—*Relative Rate of Fatigue Crack Growth*

For annealed mild steel N_S=11.6 ton. in. units for values of mean stress between 2 and 10 tons/in².

Material	0.1% proof stress tons/in²	Tensile strength tons/in²	Elongation % on 2 in.	Relative growth rate		Remarks
				Unit alternating stress	Working stress a constant proportion of 0.1% P.S.	
Austinitic Steel	16.5	43	56½	0.7	0.9	Some dependence on mean stress. Fast cyclic fracture at small crack lengths.
Mild Steel (Annealed)	15	20.5	38½	1.0	1.0	
Cold-Rolled Mild Steel	42.5	45	4	1.5	34	
Cold-Rolled Copper	14.5	20	11	4.3	3.9	
Annealed Copper	1.7	14	62	4.3	0.006	
Cold-Rolled Pure Aluminum.	5.5	7.3	14½	12.5	0.6	
Aluminum 4%—Copper alloy BS L71)	27	31	9½	30	180	Some dependence on mean stress.
Tufnol (Kite Brand, Paper Base)	—	6.1	0	55	3.5	Very approximate. Brittle fracture at small crack lengths.
Zinc	3.4	8	65½	150	1.8	Some dependence on mean stress. Fast cyclic fracture at small crack lengths.
Aluminum 5½% Zinc-Alloy Alclad (DTD 687a)	32	35	13	163	1589	Some dependence on mean stress. Initial periods of crack growth at low stresses give relative growth rates of 90 and 870.

tons/in², the fatigue cracks travelled only a short distance from the root of the notch and then stopped. Thus, however sharp the notch, these specimens showed a minimum fatigue limit of 5¾ tons/in². Frost carried out tests on specimens containing notches or cracks of different depths; he showed that, provided the notch or crack was small compared with the specimen size, the minimum fatigue limit (that is, the stress required to propagate the crack) was governed by the equation:

$$S^3 l = \text{constant}, \quad \ldots\ldots (2)$$

where S is the nominal alternating stress and l is the length of the crack, including the notch. In practice, the length of a non-propagating crack at the root of a sharp notch is usually small, compared with the depth of the notch, so that the minimum fatigue limit can be determined approximately by substituting the depth of the notch for l in equation (2). Figure 6 shows the relation between the minimum fatigue strength and notch depth, derived for the mild steel and also for a nickel-chromium steel of 65 tons/in² tensile strength and an aluminum-4.5 per cent copper alloy; all of these were tested by Frost.

Non-propagating cracks do occur in service under some circumstances as, indeed, was known many years before their presence was recognized in notched fatigue specimens, tested at constant stress amplitude in the laboratory. Such cracks are often found, for example, in locomotive axles at the press-fit with the wheel; they are a result of stress concentration or fretting corrosion. The presence of a fatigue crack in a component in service is almost always a source of danger, because the occurrence of an unusually high load may be sufficient to propagate an existing fatigue crack or even to cause complete failure by static fracture.

LOW ENDURANCES

There has been a growing interest in recent years in the fatigue resistance of metals at low endurances [16]. There are a number of applications where engineering parts are subjected to only some hundreds or thousands of stress reversals during their working life; for example, pressure vessels, the landing gear of aircraft and the working parts of guns. For such low endurances, ductile metals are able to withstand an appreciable repeated plastic deformation. Consequently, the stress is not directly proportional to the strain and it is necessary to draw a distinction between resistance to alternating stress and resistance to alternating strain. In practice, it is often the resistance to strain that is more important because fatigue failures almost always propagate from regions of stress concentration where the material is confined to a given strain range, by the surrounding material deforming elastically.

The range of strain, ε, that a material can withstand for a given endurance will be partly elastic and partly plastic, so that one may write:

$$\varepsilon = \varepsilon_e + \varepsilon_p \quad \ldots\ldots (3)$$

where ε_e and ε_p are the ranges of elastic and plas-

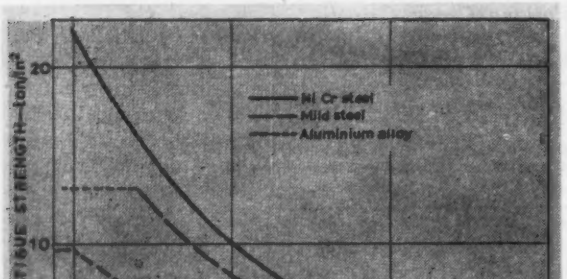

Figure 6. Relationship bewteen minimum fatigue strength and notch depth.

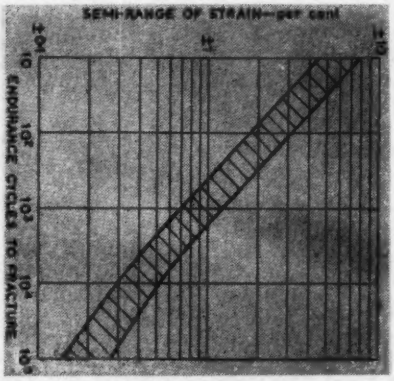

Figure 7. Resistance of ductile metals to alternating strain at low endurance: little may be gained by using higher strength materials.

tic strains, respectively. In an interesting series of experiments, Coffin [17] has shown that ε_p is approximately related to the endurance N by the relation:

$$\varepsilon_p\sqrt{N}=\text{constant} \quad \ldots\ldots (4)$$

Experimental results for a wide range of ductile metals obey this relation quite closely for endurances up to about 10^5 cycles. Moreover, the value of ε_p for failure in ¼ cycle, as computed from equation 4, agrees fairly closely with the true strain at fracture in a static tensile test. Thus, the repeated plastic strain, ε_p, a material can withstand is quite closely related to its ductility. The range of elastic strain, ε_e, it can withstand, on the other hand, is directly proportional to its fatigue strength and for short endurances this is quite closely related to its tensile strength. It is therefore to be expected that, for very short endurances when plastic strain predominates, resistance to total alternating strain will depend primarily on ductility; and it will depend primarily on strength for longer endurances, when elastic strain predominates. In fact, for quite a wide range of endurance, the strain range is almost independent of the material. This was demonstrated by Low at the Fatigue Conference [18] for a number of steels and aluminum alloys, copper, brass and phosphor-bronze and subsequently by Kooistra [19] for steels ranging from 25 to 87 tons/in² tensile strength. All the results quoted by Low and Kooistra lie within the narrow limits shown in Figure 7. The practical implication of this result is that for, short endurance applications, little may be gained by the use of high strength materials.

HIGH TEMPERATURES

The main interest in fatigue at high temperatures during the last few years has been in thermal fatigue, that is, failures caused by repeated strains due to rapid changes in temperature. It is a particularly serious problem, for example, in gas turbines, where high rates of change of temperature can occur in blades and flame tubes. It is also a problem in power stations where fatigue cracking may result from intermittent operation or from the wetting of heated metal surfaces [20] [21].

Thermal fatigue sometimes results in the formation of a network of fine cracks on the surface of a metal, known as craze-cracking. This occurs in components such as ingot moulds, hot rolls and dies, steel railway wheels, motor car brake drums and the bores of guns [22]. Fluctuations in temperature during the operation will produce thermal stresses, but experience has shown that failure is more likely to result from the much larger expansions and contractions occurring when equipment is started and stopped. This has been convincingly demonstrated, for example, by full-scale tests on a gas turbine engine [23]. By repeated starting and stopping of the engine, fatigue cracks were produced after 85 cycles in blades made of the nickel-base alloy M-252 and after 295 cycles using the cobalt-base alloy S-816. In contrast, no cracks were produced by running at full power for 360 hours, nor by rapid cycling between idling and rated speeds for 16 hours, although crack formation was accelerated when operation at rated speed and temperature was added to each start-stop cycle. These experiments provided a direct comparison of two materials under service conditions, but such tests are very expensive and most information of thermal fatigue resistance has been determined from tests in which a specimen or component is subjected to controlled heating and cooling cycles until cracks are observed [22], [24-26].

The resistance of a metal to thermal fatigue depends principally on three properties; its thermal conductivity k, its coefficient of thermal expansion α and its resistance to alternating strain ε. A high value of k is beneficial because it reduces the tem-

perature gradients, while a low value of α is desirable because the thermal strains induced are directly proportional to it. Values of α and k are known for most metals, so that a comparison of the thermal fatigue resistance of different metals can be made if ε is determined. One method for obtaining such data has been developed by Coffin [27] and is being adopted by other investigators [28-30]. This consists of a thin tubular specimen which is rigidly fixed at its end and alternately heated by a high current, and air-cooled. The fluctuating temperature produces a fluctuating axial strain in the specimen and the test is continued until the specimen fractures.

As thermal fatigue failures generally result from the starting and stopping of plant, failure is usually the result of a relatively small number of stress reversals and, in fact, it was the study of thermal fatigue that led Coffin to his study on failure at low endurances to which reference has already been made. There is some evidence that, when the temperature is fluctuating, the value of ε is about the same as if the temperature were held at the maximum all the time. In other words, the resistance to thermal fatigue may be directly related to resistance to alternating strain at the maximum temperature.

THE EFFECT OF ENVIRONMENT

There has been an increasing interest in fretting corrosion recently and its action is now more clearly understood. In the early stages of fretting, the damage is similar to the wear resulting from unidirectional sliding. As fretting proceeds, more wear particles are produced and, owing to the small relative movement, these cannot escape but build up between the two surfaces [31]. Metal to metal contact ceases at an early stage and subsequent damage results from the rubbing of the wear particles against the surfaces.

The effect of fretting corrosion on fatigue strength is much less clearly understood. Fenner and Field [32] have shown recently that the adverse effect results predominantly from the first stage of fretting, involving the welding and fracture of metal to metal surfaces. They showed that the fatigue strength of the aluminum alloy L65 was markedly reduced when pads of the same material were clamped to fatigue specimens in order to induce fretting. However, when pads of a resin-bonded aluminum oxide were used—the same oxide as is produced by metal to metal fretting of an aluminum alloy—the fatigue strength was only slightly reduced, even though the wear damage was much more severe than with metal pads. They showed further that, in the specimens with metal clamps, the fatigue damage produced by fretting, was fully developed after about one-fifth of the total endurance, for the life was not affected if the clamps were removed at this stage. This is very similar to a metal's behavior in corrosion fatigue, where it is found that the fatigue life is not significantly increased and may actually be reduced by removing the corroding liquid at a comparatively early stage in the test [33] [34]. The explanation of both the fretting and the corrosion effects is that fatigue cracks are formed at an early stage and most of the life is associated with crack propagation. Frost's work on non-propagating cracks may well lead to a better understanding of fatigue under these conditions.

The resistance to fatigue failure under fretting conditions can be increased by surface treatments such as shot peening, surface rolling and case-hardening or by the use of surface coatings such as molybdenum disulphide and P.T.F.E. [35]. However, these methods only alleviate fretting and modifications in design or fabrication may be more effective. Fretting can be eliminated, of course, if integral construction can be used and marked improvements in the fatigue strength of bolted joints can be obtained by using interference fits or very tight clamping.

The considerable improvement that can be obtained with an interference fit has been demonstrated by Low [36] in tests on aluminum alloy lugs. He found that, with a 1 in. pin, the optimum interference was about 0.007 in. and that this increased the range of pulsating fatigue strength (from zero load to a maximum load in tension) from about 1½ to 9 tons/in^2. Similar results have been reported from other sources [37] [38]. With the aid of tight clamping in a bolted joint, a considerable proportion of the load can be transmitted by friction between the plates, particularly in thin sheet. Fisher and Winkworth [39], for example, obtained an increase in fatigue strength of 4 times by tight clamping of the bolts on 16 s.w.g. (0.064 in. thick) aluminum alloy sheet. Heywood [40] has shown that it is advantageous to combine tight clamping with a very close spacing of the bolts.

By contrast with the interest in fretting corrosion, the study of corrosion fatigue has been neglected in recent years, although a number of reviews have been written on the subject [33] [41]. Corrosion fatigue was investigated very fully between the wars and it was shown that the fatigue strength of a metal in a corrosive environment depended primarily on its corrosion resistance. For, if the metal was susceptible to corrosive attack, its fatigue strength was very seriously reduced. It was found that corrosion resistant surface coatings could effectively prevent these serious reductions.

Mention might, perhaps, be made of the high resistance of titanium to corrosion fatigue. It is claimed that commercially pure titanium shows a higher fatigue strength in a 3 per cent salt solution than in air [42].

METALLURGICAL ASPECTS

A considerable effort is being devoted to the improvement of the fatigue strengths of high tensile steels. For tensile strengths up to about 80 tons/in², the fatigue strength of steels, determined on smooth polished specimens in bending, is approximately equal to half the tensile strength. Above 80 tons/in², however, the ratio between fatigue and tensile strengths is more variable. Often, no further increase in fatigue strength is achieved by increasing the tensile strength above 100 tons/in². It is thought that the reasons for this are, firstly, the stress-raising effects of inclusions and other inhomogeneities in the structure and, secondly, adverse residual stresses that are not relieved by the low tempering temperatures used to achieve high strength.

It has been shown by work at the British Iron and Steel Research Association [43] and by Cummings, Stulen and Schulte [44] that the fatigue life is inversely related to the size of the inclusion from which the fatigue fracture starts. However, Atkinson [45] has concluded in a recent paper that the size of the inclusion alone does not offer a basis for comparison, but that the number and shape of the inclusions must also be considered. On this basis he has introduced a 'Fairey' inclusion count which take account of these factors and he has shown that this can be correlated with the fatigue strength.

Owing to the influence of inclusions, the manufacturing process can considerably affect the fatigue strength of high tensile steels. This was shown by Frith [46] in a paper presented at the Fatigue Conference and some further evidence is provided in a recent paper by Fisher and Sheehan [47] for the alloy steel A.I.S.I. 4340, heat treated to tensile strengths 115 and 140 tons/in². An attempt to correlate the fatigue strength with the inclusion size was not very successful, although the latter was small in the melt with the highest fatigue strength and the melt with the largest inclusions had the lowest fatigue strength. Good results were obtained using vacuum melting but the highest fatigue strength was achieved by air induction melting, deoxidizing first with carbon and finally with aluminum. This material, after oil quenching from 1550°F (843°C) and tempering at 400°F (204°C), had a bending fatigue strength of 66 tons/in², based on 50 per cent survivals at 10^7 cycles. The authors thought that this might be the highest fatigue strength yet achieved for any metal, though this may be challenged by copper-bearing steels for which very high fatigue strengths have been claimed recently [48]. A number of investigations have been made into the influence of the microstructure of high tensile steels and these have confirmed that mixed structures have lower fatigue strengths than a structure consisting wholly of tempered martensite or of bainite [49-51].

Recent work on aluminum alloys has shown that the size and number of intermetallic particles affects the fatigue strength of a high strength aluminum-zinc-magnesium alloy in a manner similar to that of inclusions in high tensile steels [52].

BASIC ASPECTS OF FATIGUE

A better understanding of the processes underlying fatigue failure is gradually emerging from the development of improved and new techniques of optical and electron microscopy [53]. In a paper at the Fatigue Conference, Forsyth [54] showed that in an aluminum-copper alloy, fatigue stressing resulted in the extrusion of thin ribbons of metal from the surface. The effect has been subsequently observed in other metals, the size of the ribbons depending on the material and the stress conditions [55]. It has also been found that sharp crevices or intrusions are formed at the surface as a result of fatigue stressing. Forsyth [56], for example, has demonstrated this in silver chloride—which is transparent—by using transmitted light to examine the structure below the surface. Both extrusions and intrusions have also been observed with the electron microscope on replicas of the surface of copper specimens that had been subjected to alternating stress [57] [58], and by the examination of taper sections with the optical microscope [59]. It is believed that intrusions are a likely source of fatigue cracks and a number of attempts have been made to explain their formation by the movements of dislocations [60].

Perhaps the most significant outcome of the recent work with improved techniques is the detection of microfatigue cracks at an early stage in the life of the specimen. This was demonstrated by Thompson for copper and has been observed subsequently on a wide range of materials [53]. The regular way in which fatigue cracks spread has been demonstrated by Forsyth and Ryder [61] by examination of fatigue fracture surfaces with a high-power optical microscope.

INDUSTRIAL SIGNIFICANCE

Much of the impetus for research on fatigue since the war has come from the realization of its dangers in aircraft structures. One outcome of this interest is the introduction by the Royal Aeronautical Society of *Data Sheets on Fatigue* [62]. While these relate mainly to stress conditions in aircraft, they should prove useful to workers in other branches of engineering concerned with fatigue. Among the subjects so far covered, are geometric stress concentrations, fatigue data for joints and the application of statistics to the interpretation of fatigue results.

Aircraft designers are now giving more attention to the rate of propagation of fatigue cracks [63] and also to the static strength of cracked structures. It is found that the effect of a crack on the static strength is dependent on the ductility of the ma-

terial; the strength of mild steel is scarcely affected while, in aluminum alloys, fracture may occur at a stress, based on the uncracked area, of only half the tensile strength [64]. In tests on full-scale aircraft wings of the aluminum alloy 24S-T, it was found that, with 10 per cent of the tension area cracked, the static strength was reduced by 40 per cent. A progress report on this work was presented at the Fatigue Conference [65] and the full report has now been published [66].

Another branch of engineering in which fatigue research is being actively pursued is in welding and a conference on the fatigue strength of welded structures was held at Cambridge last year [67]. This will not be discussed here, except to say that appreciable improvements in the fatigue strength of welded structures are being obtained by inducing beneficial residual stresses. An article in this series dealing exclusively with welding is to be published shortly.

FUTURE DEVELOPMENTS

One can expect some further improvement in the inherent fatigue strengths of high tensile steels and aluminum alloys, resulting mainly from the closer control of inclusions. However, much greater improvements should be possible in the fatigue strengths of engineering parts, for these are often many times less than the fatigue strengths obtained on polished specimens. The improvements that are made possible by the use of interference fits of lugs and by tightly bolted joints have been quoted and the beneficial effects of surface hardening treatments are well known. It is by further development along these lines, together with close attention to detail design, that the most significant improvements in the fatigue resistance of components and structures will be achieved.

It seems certain that there will be an increased interest in crack propagation, both on a macro-scale and on a micro-scale. On the one hand, there is an increasing practical interest arising from the greater use of design for a finite life, and with it the acceptance of the risk of fatigue cracks, provided that they can be prevented from causing complete fracture. This will involve much further study of the behavior of fatigue cracks in actual components and structures. On the other hand, the discovery that micro-fatigue cracks are formed at an early stage in the fatigue life shows that a study of the rate of crack propagation is important in the fatigue process. The significance of crack propagation is perhaps not yet fully appreciated, for the present emphasis in metallographic research is directed mainly at crack initiation. However, if cracks are formed after, say, less than 10 per cent of the total life, the fatigue strength is determined predominantly by the stress required to propagate the crack.

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*RADIO-FREQUENCY SHIELDING OF CABLES**

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INTRODUCTION

THE GENERAL problem discussed in this technical memorandum can be summarized in the following way:

An end-capped sectionalized metallic cylinder contains one or more sensitive heating elements (resistors) which are connected by suitable multiconductor cabling to an external switch and DC power source. It is required that as little RF power as possible be dissipated in the load resistors. Very intense electromagnetic fields extending in frequency from below VLF through the microwave spectrum may exist in the vicinity of the cylinder and the cabling. Generally speaking, it is not feasible to

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shield the cable leads extending from the external source to the metal cylinder, but some form of shielding of the interior cables is permissible if it can be shown to be efficacious. Shielding integrity of the metallic cylinder cannot be guaranteed, and, on occasion, access doors to the cylinder may be opened for the purpose of making circuit adjustments. Various cable shielding arrangements are to be examined qualitatively for their effectiveness at all radio frequencies encountered and recommendations made concerning techniques and procedures to be employed that would be of value in minimizing radio-frequency pickup of the wiring.

The response of electric and magnetic probes within an imperfectly conducting cylinder of small radius compared to the wavelength having "shielding integrity" has been calculated [1, 2]. It is anticipated that these reports will prove to be useful adjuncts to the present memorandum concerning the minimization of radio-frequency pickup of cabling through the use of shields and other means.

FUNDAMENTAL CIRCUITS

Figure 1 illustrates a loop antenna with load resistance R_L constructed of a section of highly conducting coaxial cable. If the loop is linked by a magnetic field, i.e., a differential electric field exists between the "sides" of the structure, a voltage will be induced in the outer sheath of the cable, which appears as the voltage V across the gap shown in the drawing. This voltage is in effect applied to the sending end of a coaxial cable which is terminated in the resistance R_L. Thus the circuit portrayed by Figure 1 is a receiving antenna system and serves to demonstrate that a shield containing a break (gap) is not necessarily effective as a shield at radio frequencies.

The question may now be asked as to what steps may be taken to minimize the radio-frequency power in the resistance R_L. One obvious solution is to reduce the area enclosed by the loop by allowing the outer conductor of the coaxial transmission line to make continuous electrical contact along the conducting ground plane. The magnetic field linking the loop is then greatly reduced and the induced voltage will be very small except at the very highest frequencies. A second solution is to close the gap, thus short-circuiting the voltage V. If it is not feasible to eliminate the gap, as is sometimes the case in practice, one or more of the following steps should prove efficacious:

(a) A coaxial line filter having DC continuity may be installed in series with the cable having a cutoff frequency below the lowest frequency of the fields likely to activate the circuit. (The filter should be checked experimentally to insure no high-frequency pass bands.)
(b) The coaxial transmission line may be filled with a lossy dielectric.
(c) The inside surface of the outer sheath and the surface of the inner conductor may be coated with a lossy conductor. High-frequency currents will be forced to flow in the lossy conductor, through the operation of the phenomenon of "skin effect," and the signal will sustain increased attenuation over that available from the uncoated cable.
(d) Combinations of (a), (b) and (c).

The schemes suggested in (b) and (c) for reducing radio-frequency power in the load resistance require the use of rather long cables in terms of the wavelength, i.e., progressively longer cables are required as the frequency is decreased to achieve sufficient attenuation of the signal. In scheme (c) the thickness of the lossy metal coating must increase as the frequency decreases.

It is evident that a series resonant circuit connected across the gap will not suffice because wideband signal attenuation is required.

Even with a closed gap there is the possibility that at very low frequencies a substantial voltage will be induced into the center conductor. This is because the penetration of the fields into the metal shield increases at decreasing frequency. One can even envision cases in which the "skin depth" will actually exceed the thickness of the sheath. It is to be emphasized that shielding at very low radio frequencies is extremely difficult to execute. One possible escape is to construct the outer sheath of the coaxial cable of ferromagnetic material—the thicker the better—and accept the increased resistance of the line to the flow of DC current.

In many present-day coaxial cables the outer sheath is fabricated of copper braid. For this type of line, short-circuiting the gap does not guarantee freedom from radio-frequency pickup. It is a well-known fact that extremely high-frequency fields will pass through braid "shielding" (but with some attenuation). On the other hand, if the outer sheath of the coaxial cable consists of a copper tube and the gap is short-circuited, one is assured that no appreciable high-frequency radio energy will reach

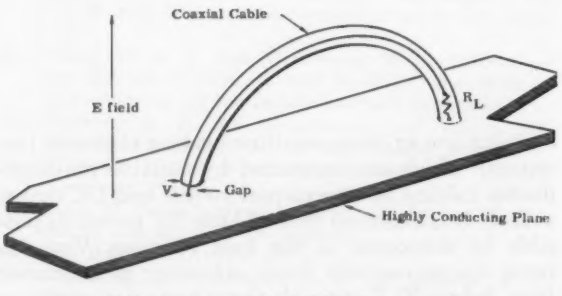

Figure 1. Coaxial Cable Loop Antenna with Load

Figure 1. Coaxial Cable Loop Antenna with Load.

the load resistor R_L at all frequencies for which the thickness of the copper sheath is many skin depths.

Figure 2 illustrates a sleeve-stub receiving antenna system. The electric field E causes a voltage V to be developed across the sending end of a coaxial cable terminated in R_L. The question of concern is how to minimize the radio-frequency pickup of the circuit. It is evident that if one ignores impedance matching, the antenna becomes progressively more effective as the frequency of the incident electric field is raised. (It is assumed that the dimensions of the sleeve-stub antenna are very small in terms of the wavelength in the VLF region.) By decreasing the dimension l, the length of the exposed inner conductor, or by extending the outer sheath of the coaxial cable, the frequency at which RF pickup becomes troublesome is increased. Even if the length of the exposed inner conductor l is decreased to zero, the coaxial cable will behave like an antenna at sufficiently high frequencies; a circular diffraction antenna without baffle (or open-ended coaxial antenna) having been evolved [3]. To prevent pickup at high frequencies when there is no exposed inner conductor, a highly conducting end cap should be used. Great care must be exercised to insure that the cap makes contact with the outer surface of the sheath all the way around its periphery. Again, the use of lossy dielectric, lossy metallic coatings, etc., in the coaxial line may prove effective at the higher frequencies to prevent RF from reaching the load resistor. At the lower frequencies the coaxial cable must be surrounded with, or be constructed of, ferromagnetic material if the most effective shielding is to be obtained.

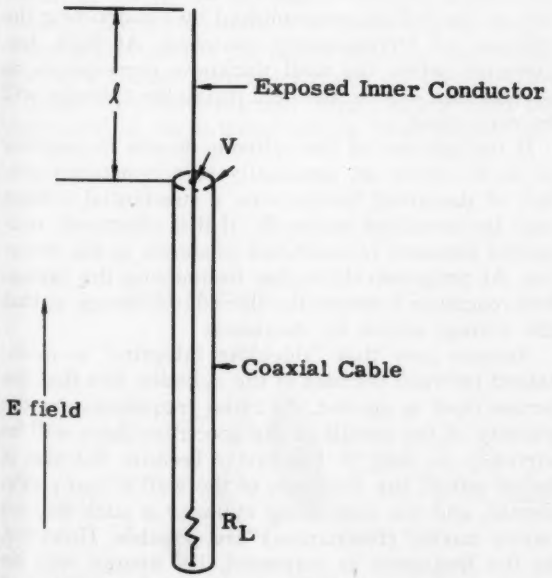

Figure 2. Sleeve-Stub Antenna Receiving System.

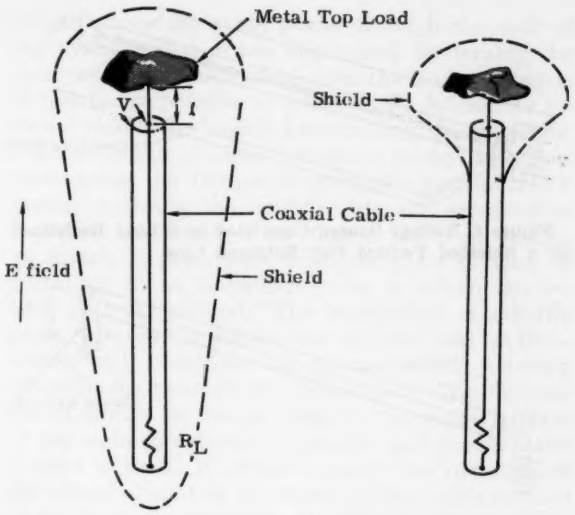

Figure 3. Top-Loaded Sleeve-Stub Antenna Receiving System.

Figure 3 shows a top-loaded sleeve-stub receiving antenna system. The operation of this circuit is the same as that set forth for the circuit without top load (Figure 2), except that for receiving systems of the same dimensions the RF pickup of the top-loaded structure is usually significant to somewhat lower frequencies than for a structure without top loading. In order to minimize the radio-frequency power in the load R_L, a closed highly conducting metal shield of suitable thickness should enclose the top load in any form suggested by the dotted lines.

Figure 4 portrays a resistance-loaded electric dipole. Even though the dimension $2l$ is quite small compared to the wavelength, a very substantial RF signal may be delivered to the load R_L. To minimize RF pickup requires the use of a highly conducting shield completely enclosing the dipole, as indicated by the dotted lines.

Figure 5 represents a circuit consisting of a battery connected to a resistor by a balanced line consisting of a shielded twisted pair. One can say cor-

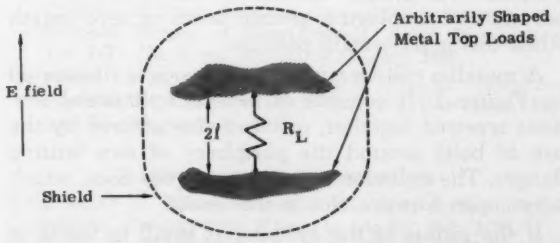

Figure 4. Resistance-Loaded Electric Dipole.

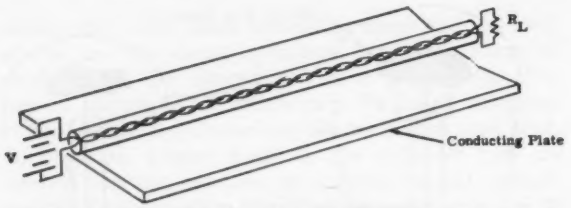

Figure 5. Voltage Source Connected to a Load Resistance by a Shielded Twisted Pair Balanced Line.

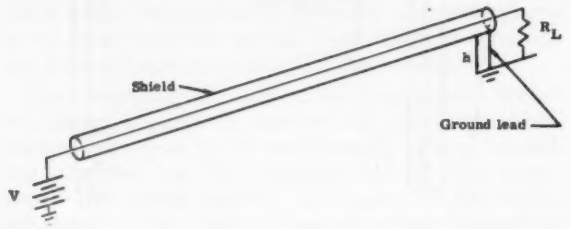

Figure 6. Circuit Employing Conventional Shielding.

rectly that as long as the line is truly balanced the RF power in the load R_L will be extremely minute if the voltage induced in the terminal "loops" by the RF field is insignificant. It is a well-known fact that if terminal effects are ignored, a line consisting of a twisted pair has lower RF pickup capability than an identical line without the twists. Furthermore, the pickup capability is further reduced drastically by positioning the line in contact with a highly conducting plane. This is brought about by image effects. The optimum arrangement, assuming that the entire circuit cannot be surrounded by a closed shield, is to encase the twisted pair in a highly conducting tube and require contact along the length of the tube with a large conducting plane.

Figure 6 portrays a circuit employing what may be described as conventional shielding, such as braid over a conducting wire. The return path is "ground." At audio frequencies this arrangement is moderately satisfactory but, as the frequency of the incident radiation increases, the length of ground lead may become significant compared to the wavelength. When this occurs, the shield and the ground lead form an inverted L-receiving antenna. The top load of the antenna, i.e., the shield, oscillates at an RF rate with respect to ground and thus serves no purpose as a shield. This circuit illustrates the importance of employing ground leads of zero length when this is physically possible.

A metallic cylinder with closed ends is illustrated by Figure 7. It consists of several cylindrical sections screwed together, or otherwise secured by the use of bolts around the periphery of two butting flanges. The cylinder contains an access door, which when open forms a slot in the shield.

If the radius of the cylinder is small in terms of the wavelength of the incident electric field, and "shielding integrity" is maintained, i.e., the access

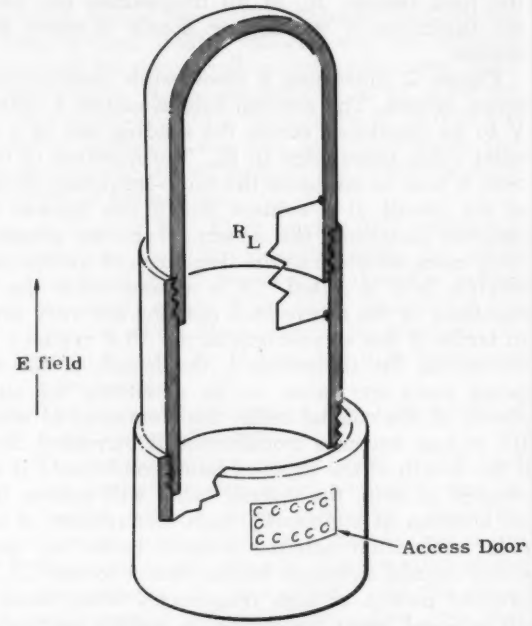

Figure 7. Sectionalized End-Capped Metallic Cylinder with an Access Door (slot).

door is closed and makes good contact with the cylinder around its perimeter, and the several sections provide continuity at high frequencies without potential drop, one can calculate the electric field inside the cylinder in terms of the incident field or in terms of the actual field existing on the outside surface of the cylinder [2]. At low radio frequencies this field may be substantial in cylinders fabricated of rather thin-walled copper or aluminum. A reduction in the field is accomplished by constructing the cylinder of ferromagnetic material. At high frequencies, when the wall thickness corresponds to several skin depths, the field inside the cylinder will be very small.

If the sections of the cylinder do not fit together so as to insure an essentially zero resistance contact at the lower frequencies, a substantial voltage may be developed across R_L, if it is effectively connected between two sections as shown in the drawing. At progressively higher frequencies the capacitive reactance between the threads decreases, so that the voltage across R_L decreases.

Assume now that "shielding integrity" is maintained between sections of the cylinder, but that the access door is opened. At radio frequencies in the vicinity of the middle of the spectrum there will be virtually no field in the cavity because the slot is below cutoff, the thickness of the wall is many skin depths, and the size of the cylinder is such that no cavity modes (resonances) are possible. However, as the frequency is increased, RF energy will be admitted to the cavity through the phenomenon of cavity resonance.

It would appear that the difficulty can be "postponed" to higher frequences by reducing the dimensions of the access door and by compartmentalization of the cylinder. The idea is to keep the slot below cutoff as high in the RF spectrum as possible and substitute for a large cavity resonator a number of smaller ones. The compartmentalization of the cyclinder must be executed with care, allowing no "floating" contacts between baffle plates and the inside surface of the cylinder, etc.

Another way of reducing the field within the cylinder is to fill it with a broadband RF absorbing material. This treatment is likely to be effective only at frequencies in the microwave region.

It is evident that reduction of the field within the cylinder is essential since all exposed wiring will pick up RF in proportion to the existing field strength.

COMPOSITE CIRCUIT

Figure 8 represents a composite circuit which violates most of the rules of shielding. Steps to be taken to reduce RF pickup by the cables will be enumerated.

Assume at the outset that it is not feasible to shield the cabling completely. Where the external shield ends, one should be sure to group as closely as possible the wires going to the battery and ground so that a loop antenna is not formed. Each conductor in this bundle should be surrounded by a dielectric of high loss at radio frequencies. (This treatment should extend the full length of the cable, i,e., from the battery to the load resistor R_L.) The entire cable should be run as close to a large highly conducting plane as possible. The greater the ratio of the lengths of shielded to unshielded cable the better. It should be grounded to the conducting plane all along its length. The lower end of the external shield should be faired out and thoroughly grounded to the external surface of the metallic cylinder. Similiar treatment should be accorded the upper end of the internal shield. Thus gaps in the shield where the cable passes through the wall of the cylindrical case are eliminated. Preferably the internal shield should close over the load resistance. If this is not possible to execute, the longer the internal shield the better. Lengthening the shield for a given length of inner conductor pushes to higher frequencies the frequency at which pickup of RF energy entering the cylinder via the access door becomes significant. Note that a balanced circuit, as shown, is preferable, from the point of view of shielding, to an unbalanced one in which the return path is "ground." The installation of a baffle plate between the access door and the load, as illustrated, will eliminate the pickup of RF entering the cylinder through the access door. The internal shield should be run as close to the inside surface of the metallic cylinder as possible and should make contact with it. If contact cannot be maintained, the shield should be grounded to the inside surface of the case at frequent points along its length, using grounding leads that are as short as possible. The loop formed in the conductors at the load resistance should have as small an area as possible. If an absolutely noninductive capacitor is available, it may be connected directly in parallel with R_L to reduce the RF drop across the load.

The best internal shielding consists of conduit. If the load resistance, R_L, is located in a box, the box should be constructed of highly conducting material (or at least be metal plated) and the conduit carefully sealed electrically to the box. The conduit must make perfect contact around its circumference with the cylindrical case at the exit hole. Cable having a braided shield, although undesirable, would probably be required outside the cylinder. The upper end of this shielded cable might be run into a metal junction box, where installation of an RF filter for each circuit involved has been made. In this way the RF pickup of the unshielded portion of cable might be reduced to acceptable values before it is transmitted through the cable to the load resistance.

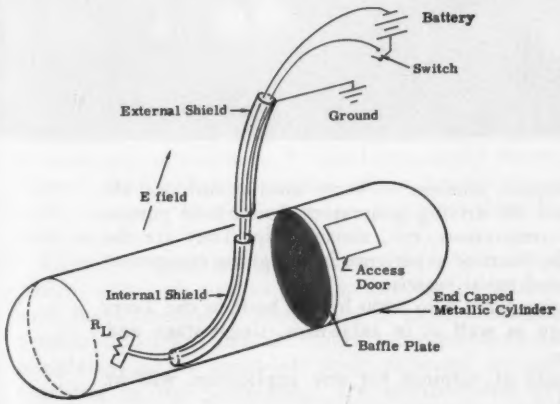

Figure 8. Composite Circuit.

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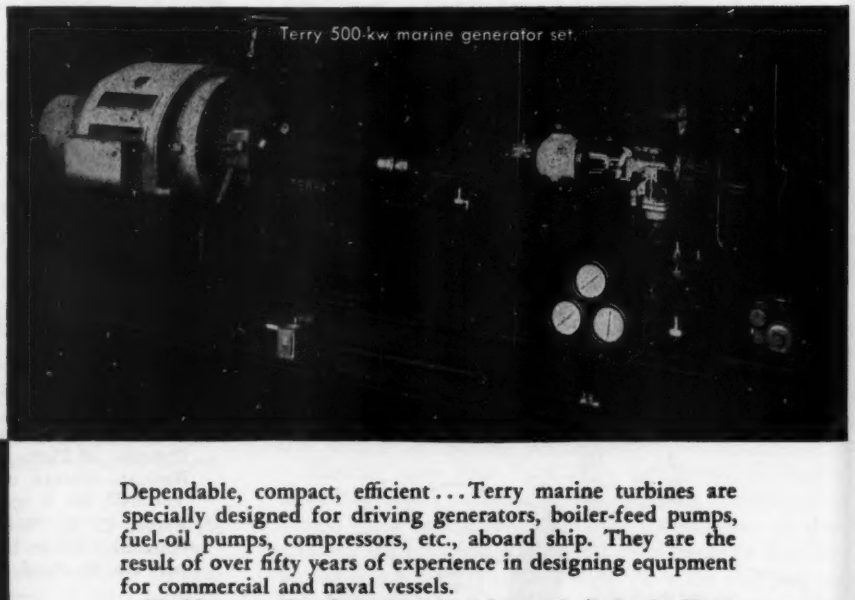
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"SHIPBUILDING AND SHIPPING RECORD"

TECHNICAL REVOLUTION IN SHIPBUILDING

ACKNOWLEDGMENT

This staff article appeared in the February 23, 1961 issue of "Shipbuilding and Shipping Record" from which it is reprinted.

TECHNICAL PROGRESS in design and building methods is of first importance in the competitive position of shipbuilders. Shipbuilding techniques have undergone a spectacular change in the past 20 years, and this is largely associated with the introduction of welding for joining steel plates. Technical evolution is usually protracted and the first British all-welded vessel was built in 1920, yet it was the demands of war for rapid ship construction that led to wide adoption of welding in the U.S.A. Despite a number of failures, the war-time mass-welded vessels were very successful, and the improvements in welding which have taken place since the war have led to the virtual cessation of riveting, except for specialized vessels. This change would not have taken place unless shipowners were convinced of its economic advantages. It has been amply demonstrated that reliable welded ships are produced more cheaply, quickly, and use less fuel than riveted vessels.

There is another development in shipbuilding, which, although not as spectacular as the change-over from riveting to welding is of importance. The production of full-size drawings, scrieve-boards and templates on the mold-loft floor, followed by plate marking in the plate-shop, are cumbersome procedures, wasteful of time, money and skilled labor. In 1944, a less costly and more elegant method was introduced. Optical marking and 1/10th scale lofting in their present form were first used in the Stülcken yard in 1951, and have since been installed in about 100 shipyards.

In October, 1960, there were about 27,700,000 tons d.w. of shipping on order in the western world. Of these, some 11,000,000 tons, or 41 per cent, are being, or will be, constructed with the use of optical equipment of G.A.G. manufacture. Despite the present recession in world shipbuilding, the number of yards using optical methods is rapidly increasing, and the mold-loft will probably disappear within the next decade or so.

The introduction and use of optical methods is only possible if it gives well-constructed, reliable, cheaper ships, more quickly. It is difficult to obtain direct comparisons of costs between yards, because of differences in constructional methods and accounting procedures. Even so, it is reasonable to assume that a trained man will do a particular job, in approximately the same time, whether he be a Swede in Gothenburg, a Dutchman in Rotterdam, or a Geordie on the North East Coast. It is the number of man hours taken for all the operations in building a ship which determines its price and delivery.

Optical marking and 1/10th scale lofting, like full scale lofting and template marking, are operations depending on skilled workers. But those who have experience of optical marking and 1/10th scale lofting are convinced that the accuracy obtained is generally superior to that obtained in a full-size mold-loft. The U.S. Navy probably exacts as high a standard of shipbuilding as any in the world, and many U.S. warships are built by optical marking.

GREATER ACCURACY

The accuracy of optical marking is due to the stability of the negatives under changes of climatic conditions (in contrast to movement of mold-loft floors and wood templates), and the ease with which the projector can be focused to give the exact distance between any two check points on the plate. It is impossible to give the savings which shipbuilders make because of greater accuracy of optical marking. However, continental shipbuilders maintain that fairing on the berth is reduced and that more accurate plates give better and more rapid welding. Herr Koops, of Lubecker Flenderwerft, states that, for a 25,000 tons cargo ship, optical marking saves about £15,000 in labor and materials over conventional methods. This figure was obtained by comparing a number of ships constructed by both methods, but Mr. Koops does not give the percentage due to greater accuracy. Other figures give savings achieved by optical marking alone on a ship of this size, of between £10,000 and £15,000 not including overheads. This does not include constructional savings on the berth. The Schlieker yard in Hamburg, uses the 1/10th scale system, efficient material handling and automatic gas-cutting machines, and has an output of 48,000 DM. per man/year, as opposed to the National German average of 28,864 DM.

SAVINGS IN SCRAP

The greatest single saving from the use of optical methods is the reduction in scrap. Kriepke, naval architect of G.A.G.,[1] says:

"The saving in scrap is greatest with transverse plates in double bottoms, and with bilge plates. Here brackets and other small items can be nested into the lightening holes; in these cases, the utilization is nearly 100 per cent. It is easier for the optical detailer working on 1/10th scale, to make the best use of a drawing than for the plater, working full-size in his shop, to make full use of a plate. Optical marking reduces scrap from an average of 10 per cent to between 21 per cent and 31 per cent."*

This statement requires examination and qualification. The use of standard sized plates is more common on the Continent than in Britain. The British shipbuilders tends to order more plates cut to size by the steel mills. The savings by this method are somewhat illusory, as the steel maker is left with scrap, for which the shipbuilder has to pay, admittedly without transport charges. This method of ordering is economic when scrap is high, but costly when it is low as with 1/10th scale plate lay-out.

We will now imagine a medium-sized shipyard, producing about 8 to 12 ships a year of all types and sizes, the steel throughput will be about 60,000 tons a year, of which about 40,000 tons will be plate and 20,000 tons angles and frames. If we assume a reduction in plate scrap from 8 per cent to 4 per cent as a result of using optical methods there will be a saving of 1,600 tons. The average price of steel for shipbuilding is a little over £40 per ton, and scrap fetches something under £10 per ton. The savings are £30 per ton on 1,600 tons, amounting to £48,000.

British loftsmen are said to be more highly skilled and more efficient than on the Continent and that the savings to be achieved here by optical marking are therefore small compared with the rest of Europe. The number of loftsmen in the U.K. is often less than half the number employed in a similar yard on the Continent. But this is due to different methods of work, and the smaller number of men in the mold-loft in the U.K. is offset by more markers, etc., in the plate shop. On the Continent, there may be 25 men in the loft and five on the shop floor, compared with five men in the loft and 25 on the shop floor in this country. The savings with the 1/10th scale system are approximately the same.

AUTOMATIC GAS CUTTING

A number of systems and techniques eliminate all or some of the mold-loft operations. The optically-controlled automatic gas-cutting machine with projector table gives the most economical production of some 20 to 40 per cent of all the plates required for a ship.[2] But it is suitable for the more curved shapes, particularly if the pieces are large. If small and complex pieces are required from a large plate, the time taken to stop and start the machine for each piece is excessive, and if bridges are used, the amount of scrap (15-17 per cent) is too large. Further, many of the shell plates in the fore and aft line cannot be automatically burnt out because accuracy is essential. These plates can only be cut by one torch because they have non-parallel side edges, and this causes distortion due to unequal heating. Practice has shown distortion up to 8 mm. in long plates. With the latest machines fitted with a projector table some of these difficulties are eliminated.

For fine ships, only 20 to 25 per cent of all plates can be cut by automatic burning machines, but this is increased to 40 per cent or even more for ships with long parallel 'midship sections, such as tankers or ore carriers.

[1] Gesellschaft Fur Anzeichen-Gerate M.B.H. Hamburg-Lokstedt, Grandweg 56-58.

* This is as originally printed. (*ASNE Editor*)

[2] Koops and Bentfeldt—The application of Flame-Cutting to Ship Construction, Schweissen und Schneiden, January 12, 1960.

It is fairly common practice to combine the Sicomat and Telerex automatic gas-cutting machine with optical marking, but with the Monopol machine the 1/100th scale negatives cannot be used for optical marking. Costly equipment such as an automatic gas-cutting machine must be kept fully employed to cover its cost. If the shipbuilder has machines for 40 per cent of the plates of the ships he is building, they will be fully employed for tankers and ore carriers, but standing idle part of the time if he is building passenger or fast cargo ships. If, however, he has only capacity for 20 per cent of the worked plate, costs will be high when he is building tankers. The best economic solution is to have automatic gas-cutting machines for about 30 per cent of the plate output. It is then usually possible to keep the machines fully occupied by mixing the plate requirement for cargo ships and tankers. This is much easier if the negatives for optical marking and the automatic gas-cutting machines are interchangeable. The G.A.G. optical system and the use of the G.A.G. projector table as fitted to the Messer Sicomat and Telerex machines, is the only combination which allows this essential interchange of negatives between optical marking and automatic gas-cutting machines.

Yards which have been operating with automatic gas-cutting machines, reading direct from 1/10th scale drawings or from 1/100th scale negatives, are changing to optical marking and projector tables because of the flexibility and economic production provided by the interchangeability of negatives, and further, the 1/10th scale system and optical marking form the only widely accepted method for making templates without recourse to the full-sized mold-loft. Automatic gas-cutting machines are sometimes used to mark plates but the cutting speed is far too slow, and capital cost too high, to make it an economic proposition.

The computer-controlled automatic gas-cutting machine, like its optically-controlled counterpart, and for the same reason, will only be required to cut 20 to 40 per cent of the plates which go to make a ship. Like the Monopol, it suffers from the disadvantage that its control is not interchangeable with optical marking. The production of the magnetic control tape by computer involves making 1/10th scale sketch drawings from which the co-ordinates of the various parts on the drawings are obtained. This information is transferred to the computer by a mathematician and typist. Little experience with computer-controlled cutting machines is yet available, but it seems unlikely that the production of 1/10th scale sketch drawings, and the transfer of the co-ordinates to the computer, will be cheaper than making accurate 1/10th scale drawings and negatives. Also, the capital cost of the computer control is greater.

PORTAL MACHINE ECONOMICAL

In every ship there are rectangular plates, which can be cut without marking, on a Portal type edge planing and cutting machine. This type of machine is cheap, both to buy, and to operate. Parallel edges and even slow curves can be cut from the roughest of sketches. As much work as possible will, therefore be done on the Portal type machine. The amount of plate which can be handled by it varies from ship to ship. There is, however, some flexibility of operation because any plates which can be cut on the Portal machine can also be cut by the fully-automatic machine.

In general, about 25 per cent of the total plate tonnage can be dealt with by the Portal machine, and, as already stated, about 30 per cent by the fully-automatic machine. Optical marking, is therefore, required for some 45 per cent of the total plate tonnage of a ship.

The weight distribution of plate in the yard is therefore:

45 per cent by optical marking	=	18,000 tons
25 per cent by Portal type planing and flame-cutting machine	=	10,000 tons
30 per cent by fully-automatic flame-cutting machine	=	12,000 tons
Total		40,000 tons

If we assume an average plate of 40 ft. by 8 ft. by ½ in. thick, the weight per plate is about 3 tons.

Two automatic gas-cutting machines, each fitted with four bays, two for cutting, and two for loading and unloading, can cut about 12 plates a day if two cutting heads are fitted, or about 18 plates a day if four cutting heads are fitted and can be used. Thus:

12 plates per day at 3 tons ..... = 8,640 tons/year

There is, therefore a shortage of cutting capacity of 3,400 tons, which can to some extent be made up by fitting four heads, and by overtime and shift working. Because of the simple nature of the cuts, the capacity of the Portal type machine will be in excess of 10,000 tons per year, and in some ships, the shortage of cutting capacity of the automatic machine can be made up by the Portal machine.

The tonnage figure for the automatic gas-cutting machines given will be fairly accurate because the plates cut lie mainly within the body of the ship and are generally only ½ in. thick. The thickness of the plates cut by the Portal machine will vary widely.

Continental experience shows that it takes about half an hour to mark a plate by optical projection. British practice agrees fairly well, and it is reasonable to accept a figure of 35 plates a day. Thus:

35 plates per day at 3 tons ..... = 25,200 tons/year

Two projectors, therefore, have an excess plate capacity of 7,200 tons. This excess capacity will be used for frame and bending roll templates and for

any shortage from the fully automatic gas-cutting machine.

This imaginary shipyard will probably meet its steel requirements with two fully-automatic gas-cutting machines, two optical marking projectors for plates and templates, and one Portal type edge planing and cutting machine.

The man-hours worked in the optical detailing office are about 50 per cent of those worked in the mold loft. The man-hours worked for optical marking are 35-50 per cent of those taken with the mold-loft procedure for comparable plates. If due allowance is made for the greater skill claimed for British loftsmen, it seems reasonable to assume a minimum saving in marking time of not less than 50 per cent.

MANPOWER SAVINGS

Figures obtained from British yards show that the mold-loft man-hours for a throughput of 60,000 tons a year of steel are about 40,000, and the marking times about 105,000. This is based on no repeat ships. Direct labor costs, including the cost of template material are about 6s. 2d. per hour, giving a total cost of £44,700. Overheads have not been included in the above figure, but there will be a direct saving in labor costs resulting from the installation of the 1/10th scale system of £22,350. This takes no account of the reduction in the number of plates to be marked, as a result of the installation of Portal and fully-automatic gas-cutting machines.

Information obtained from a yard in Germany shows that 2.17 man-hours are required per ton of steel for 1/10th scale drawing and optical marking and template manufacture. This compares favorably with the 3.73 man-hours per ton required by the full-scale mold-loft processes used in the U.K. The difference is equal to 1.56 man-hours per ton for 60,000 tons of steel at 6s. 2d. per man-hour, *i.e.* £28,800.

The total annual savings, exclusive of overheads, are:

From scrap	=	£48,000
Direct labor charges	=	£22,350 to £28,800
		£70,350 to £76,800

The cost of the equipment is approximately as follows:

2 Optical projectors	£ 6,000
1 Camera	1,600
Drawing office and photographic equipment	2,300
1 Projector tower	1,000 or 4,800
2 Marking tables	2,400
Platforms, ramps, etc.	600
Constructional, and electrical work, erection, etc., approx.	£10,000 or £15,000
Total for optical marking	£23,900 or £32,700

(Note: The lower figure applies if the tower can be built in the plate shop roof. The higher figure applies if a separate rigid tower has to be constructed.)

2 Automatic gas-cutting machines	£36,000
1 Portal edge planing and cutting machine	6,000
2 Projector tables	4,200
Total for gas-cutting machines	£46,200
Total	£70,100 or £78,900

If we allow depreciation at the rate of 10 per cent per annum, and interest charges of approximately 8 per cent the savings are reduced by £14,000, and, therefore, become £56,100 to £65,900 a year. This does not include savings on overheads and the reduction in the number of men employed as burners, etc., because of the increased efficiency of the Portal-type machine and the fully-automatic gas-cuting equipment.

It is, therefore, probably true to say that the equipment will pay for itself in a little over one year, and that, as a result of its use, there will be a saving of between £60,000 to £100,000 per annum thereafter.

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LESLIE E. ALSAGER

SUBMARINE DESIGN— A MULTITUDE OF COMPLEX PROBLEMS

THE AUTHOR

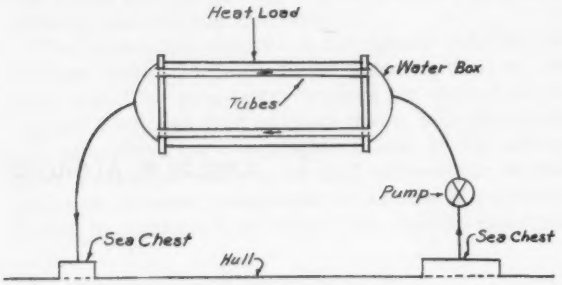
is a native of Minneapolis, Minnesota. He received the Bachelor of Mechanical Engineering degree from the University of Minnesota in 1949 and in 1953 became a Registered Professional Engineer in that state. He has been a civilian Marine Engineer in the Scientific and Research Section of the Machinery Branch of the Bureau of Ships since 1951 and with the Bureau since 1949.

THE NAVY HAS for generations been interested in the military capabilities of the submersible. With the advent of nuclear power the potentialities of the submersible have increased tremendously. Thus industry, including the previous non-participant in the submersible engineering field, is becoming increasingly interested. Many concepts of tasks for the submersible are being evolved by industry in an effort to "get their foot into the door." Many propulsion cycles are being evolved. Some are intriguing. Others become or are repetitive. Many cycles depend on aircraft practice of extremely light weight equipment with this equipment operating in an air medium, this air medium having a density and characteristic unlike the salt water in which a military submersible must operate to be effective. The field of submersible propulsion cannot be summed up like a text book. Nor can an all inclusive handbook be written as an absolute guide. Submersible engineering requires real compromises of space, weight, material selection and performance. Performance becomes a criterion when it is defined by the characteristics of the other three plus the noise parameters. Thus this paper is not all inclusive but is intended to give to those "non-submariners" through selection of a single component and its associated system a limited reflection of the problems, the engineering, and the compromises which are involved. To portray this reflection, the author has chosen to take a common heat exchanger served by sea water and develop several systems all basically serving this heat exchanger but each slightly different for different reasons. It is not the author's intention to state that these are cycles which the Navy has used, will use or intends to use. Rather the systems are described simply for portrayal of submersible design considerations.

DIRECT CONNECTED SYSTEM (SIMPLE DESIGN)

In engineering parlance, a direct connected heat exchanger system would involve a piece of pipe carrying water to the heat exchanger and another piece of pipe carrying this water away from the heat exchanger. Somewhere in the piping would be a

pump to provide the circulation head to the salt water. This pump would be designed for friction and head loss of the system. This to the engineer becomes simple enough, probably elementary. A diagrammatic of this simple system is shown thus:



The problems of design are not apparent from the diagrammatic. To develop these problem areas in a manner which would provide separation of each is rather difficult. Thus, while each probably can be categorized there will be a discussion of each which will be of necessity intertwined with the others.

The designer using sea water must recognize the corrosive-erosive nature of the fluid. Material selection becomes an immediate dependent parameter to the system. Most equipment designers are acquainted with the basic rudiments of material selection. However, a limited number become involved in material selection for sea water usage. Sea water basically contains NaCl. Unfortunately, it also contains CO_2, CO and most of the other gases of the atmosphere plus sand, sea life (fish and organism) as well as flotsam or just plain trash. Thus, material selection becomes first a corrosive problem and secondly an erosive problem. Next, it must be recognized that this material is in piping which is attached to the hull. Thus making the hypothesis that the heat exchanger is rigidly fixed to the hull the piping must absorb the strains imposed on it by deflection of the hull inward as the submersible goes deeper or the water gets colder. This deflection can be likened to that which happens when one squeezes a metal can. The pressure on the submersible, of course, comes from the sea water pressure outside due to the head of water above the submersible. An example of this pressure can be simply shown by stating that for each 225 feet of depth a pressure of 100 pounds per square inch would be exerted on the hull. Thus the piping which sees sea water pressure also must be designed for the hoop stress in the pipe as well as the strains imposed upon it by the hull deflections. These principal stresses cannot be overlooked. Neither should the fatigue quality of the material be overlooked since the number of these pressure and temperature cycles could be significant. While the calculated principal stresses become subject to engineering solution the behavior of the material in sea water under fatigue criteria becomes an engineering headache. Most published data on fatigue is given for a rotating specimen under unilateral stress operating in air. Corrosion fatigue in sea water (which is what the designer is interested in) reacts differently. Secondly, the necessary data must be based on long time "slow fatigue" tests. The reason for this is that most materials, particularly ferrous materials exhibit a marked decrease in endurance limit under this condition. (While the author doesn't want to get into a detail materials discussion, he would like to state that a two-to-one reduction in endurance limit is not unusual.)

Previously, it was mentioned that the sea abounds with entrained foreign material as well as the salt. This predisposes then an erosion problem which becomes dependent upon the velocity of the fluid and the configuration of the system as well as the roughness of the piping. It is axiomatic that for a fixed delivery of gallons per time of fluid flowing the higher the velocity in the pipe (in subsonic range) the smaller the pipe. Since the foregoing discussion on deflection strains could reflect a flexible (sprawling) piping design similar to that of high temperature power piping, a decrease in outside diameter of the piping could reduce the run of piping for any fixed depth. High velocities are an important consideration. Unfortunately, high velocities have a direct relation to erosion. Particularly where there are obstructions or bends in the piping and turbulence occurs, erosion impingement at a point in the piping may be found. Designs of moderate velocity, where obstructions are involved, have literally eaten away the material in a very short time period. One method of combating this, of course, is to provide a strainer in the system. The mesh size of the strainer would be dependent on the size of the obstructions required to be removed. Where large quantities of water are flowing and flow velocities are moderate, it is usually economically wise to make this strainer mesh size (hole size) slightly smaller than the inside diameter of the tubes in the heat exchanger. Thus, anything that gets through the strainer can be assumed to be of a size that will pass through the heat exchanger tube. This minimizes the possibility of the heat exchanger with its small tubes acting as the strainer. The suction sea chest shown in the diagrammatic can often contain this strainer, the strainer being nothing more than a perforated plate across the opening. Any strainer in the system should contain a hole area of greater dimension than the area of the system it serves so that inherent partial clogging of the strainer will not materially affect the system performance of the heat exchanger through insufficient cooling water. Fish and sea weed are two of the most common items found in ship sea water strainers.

Because it is unwise to send a diver overboard

to clean out this sea chest strainer each time it becomes fouled, a system must be evolved to backflush the sea chest to get rid of the collected debris. Thus, this simple heat exchanger circuit may now have an auxiliary system to provide this backflush. This auxiliary system can be of the liquid or air type, but must furnish the backflushing fluid at a pressure above that of the head of the sea water over the strainer.

Previously, the ramifications of hull deflection relating to pipe strains was discussed. Obviously reaction forces at the heat exchanger and at the sea chest must also be considered. At the same time the thermal expansion of the heat exchanger due to the temperature of the medium being cooled must be included. In the schematic shown, the thermal growth of the heat exchanger will increase the reaction forces and the strains to the piping. The heat exchanger is oftentimes mounted on one fixed and three sliding feet. The mounting scheme or feet arrangement must be selected on the basis of minimum strains and reactions to the piping and to the heat exchanger. All foregoing considerations of the material selection for the piping thus become important to the heat exchanger design also.

Sea water exhibits a deposition characteristic when associated with temperature. Salting of a sea water heat exchanger oftentimes plays a finite role in selection of the characteristics of the heat exchanger. Where high temperatures or hot spots are expected, a non-salt water intermediate heat exchanger, i.e. fresh water for example, should be considered.

Dead ending of systems must be avoided due to organism and marine growth always present where no flow is experienced. This marine growth can literally choke a system making the system nonuseable until the obstruction is cleared. Organisms attaching to the surface of the dead ended system form corrosion pockets or corrosion cells as well as possibly causing turbulence with consequent impingement erosion to the downstream surface when the system is used.

While little additional can be said about the choice of the fluid circulating pump for the system, it must be compatible from the material selection standpoint with that of the rest of the system to which it is matched and connected.

Nowhere in this paper has the compatibility of materials to each other been mentioned. This is an important point and cannot be over-looked. Sea water can be likened to an electrolyte where dissimilar materials connected to each other or in close proximity to one another form an electrical cell with consequent emission into the solution of the less noble materials. This is a knotty problem but generally can be aborted through the following:

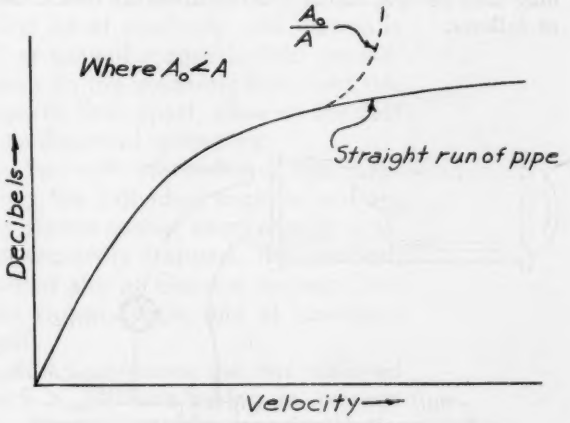
a. Use of non-dissimilar materials

b. Use of waster pieces

c. Recognition of the problem and establishment of a rigid maintenance replacement schedule.

d. Acceptance of a short life criteria.

The foregoing involves the mechanics of design from a physical standpoint and basically affects the performance and reliability characteristics of the system. Oftentimes this is all that the system needs to satisfy. However, where the system must be designed to be a non noise radiating system, additional criteria must be satisfied. Air-borne noise emanating from the system plays a relatively minor role in the design of the system except where it becomes objectionable to the people working in the area. It does play a part in the aspect of noise radiated to the sea. In this simple system, there are flow noises occurring as a result of turbulence of the fluid, pump noise as a result of vane excitations and fundamental rotation excitation of the pump unit. From a flow noise standpoint, it can be generalized that the higher the flow velocity the higher the noise level in the system. A simple curve of this can be shown. Should a restriction or discontinuity such as a flow balance orifice or valve be placed into the system, this noise level could show a marked increase.



VELOCITY VS. DECIBELS

In the simple cycle, a noise abatement program (that is restricting the level of the noise which gets into the water outside of the submersible) may involve many equipment and design compromises. For example, fluid filters on either side of the pump tuned to the cutting frequency of the vanes or to the rotational frequency of the pump may be applicable. These filters can be likened to a muffler. Further, a decrease in flow velocities and minimization of obstructions may be necessary. Pump isolators which interrupt the metallic path through which the noise would normally travel would be required. Should these isolators be a soft mount or pad under the base of the pump, then flexible cou-

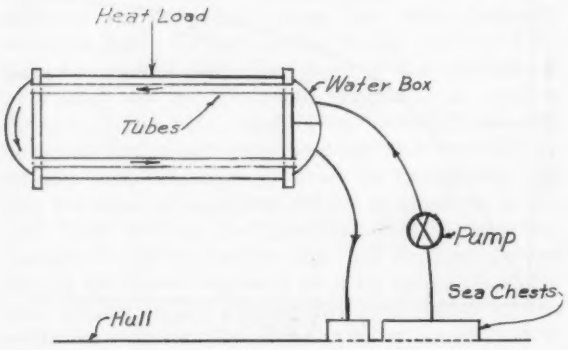
plings in the piping must be installed to absorb the movements of the pump occurring from:

(a) Thrust forces caused by submergence pressure acting on the volute of the pump.
(b) Vibrations to the pump caused by external hull excitations.
(c) Vibrations of the pump caused by internal pump frequency excitations.

From the above, it can be deduced that this simple system, when radiated noise becomes important, becomes extremely complex and subject to many design compromises.

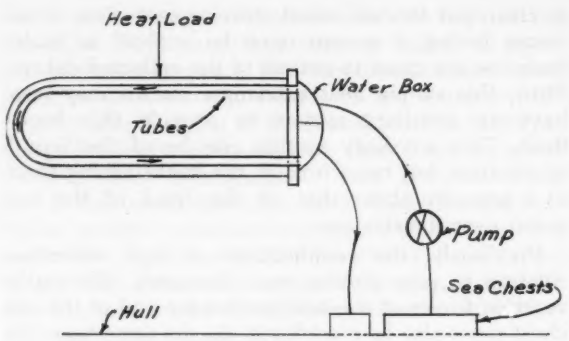
OTHER CYCLES OR SYSTEMS

A marine engineer in the submersible design field rarely encounters the simple cycle or can use or wants to use such a cycle. Too many indeterminates of design are involved which makes the compromises disparaging. Thus he attempts to provide cycles or systems which give to him the requirement he must match. For example, reaction forces on the heat exchanger may become important. Thus he may go to a two pass arrangement where the inlet pipe and the outlet pipe are on the same end of the heat exchanger. This two pass arrangement may take several forms. Two of these are described as follows:



a. Return water box design

This arrangement permits the heat exchanger to be rigidly fixed or anchored at the pipe end. Thus the thermal growth of the heat exchanger with its previous effect on piping strains and reactions is minimized. Further, a design such as this (assuming the same overall length of the simple heat exchanger) provides for an increase in the heat transfer surface and a reduction in the circulating water required for cooling. This helps alleviate some of the problems of the simple cycle of pipe size, runs of piping, etc. These effects must be balanced against the increase in pump head (system resistance) and the greater diameter of this two pass heat exchanger.



b. U-type arrangement

Oftentimes where space, weight or tube removal become important, a "U tube arrangement" may be considered. This arrangement differs from the previous two pass installation in that the tubing is continuous and is configured in a "U-bend".

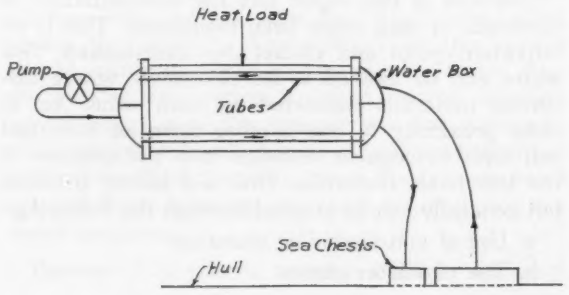
This installation provides the advantages of the previous double pass unit plus:

(a) Increased surface heat transfer area
(b) Ability to remove the tube bundle as a unit for maintenance.
(c) Decreased circulating water quantity for the same flow velocity in the tubes.
(d) Elimination of one water box

Disadvantages of the unit are:

(a) Possible tube plugging as a result of obstructions lodging in the U bend.
(b) Isolation of a leaking tube approximately doubles the surface heat transfer area lost.
(c) Increased turbulence in the bend.

Non economical noise isolation of the circulating pump as mentioned in the simple cycle may require some other arrangement of design and further compromises. For example, it is theoretically possible to use each pass of the two pass heat exchanger as a noise trap (or muffler) of the pump excitation noises by installating the circulating pump at the return end of the heat exchanger either on the return water box or in a pipe configuration at the return water box end.



PUMP ON RETURN WATER BOX END

Disadvantages of this arrangement involve the physical installation and the isolation of the metallic path through which the noise can travel. Further, the muffler (each heat exchanger pass) must be tuned to the frequency or frequencies which the installation is attempting to abate. Compromise of design from a performance standpoint have to be assessed against this tuning requirement. Pump characteristics particularly the cavitation index must be reckoned with because of the frictional resistance of the heat exchanger and piping on the suction side of the pump.

CONCLUSIONS

The foregoing does not provide an absolute solution to a sea water cooled heat exchanger system. It is believed, however, that it does give to the engineer entering the submersible field a gleaning of the problems encountered. It must be pointed out that this discussion involved a heat exchanger unit in its simplest form. Many other parameters of design are affected depending upon the installation concept and the type of fluid which is being cooled by the sea water. Further, it is hoped that those considering this field as an avocation recognize that a submarine at sea represents to the submarine designer a multitude of complex engineering problems, many of which are more complex than this heat exchanger example.

CUSPED MAGNETIC FIELDS USED IN THERMONUCLEAR RESEARCH

A relatively new approach to the thermonuclear problem is reported in an article by F. L. Ribe of Los Alamos on "AEC-Sponsored Controlled Thermonuclear Research" in the British journal ***Nuclear Engineering*** for October, 1960. It involves injecting a dense stream of energetic plasma from a gun into a magnetic bottle in such a way as to "inflate" it.

In the entropy-trapping concept a fast jet of relatively cold plasma is projected axially into a "picket-fence" or cusped magnetic-field geometry. The plasma exerts a dynamic pressure on the magnetic field, and this is made great enough to force the magnetic lines apart, allowing the fast jet to enter the central region of the confinement geometry.

The action of forcing its way in, together with scattering of the beam off the inside magnetic walls, randomizes the individual particle motions so that, on turning the jet off, the inside plasma cannot exert enough pressure to force its way out and it is consequently trapped. This method, originated at Los Alamos, is being studied also at General Atomic Division of General Dynamics Corporation in San Diego, and at Lawrence Radiation Laboratory in Livermore, Calif.

Various types of plasma guns are used: A Los Alamos gun has achieved a stream of hydrogen plasma of some 5×10^{16} ions having an average energy of the order of 10 kev. In the General Atomic experiment the plasma is observed to open the input throat of the cusp and to be detained inside for about 40 microsec with a number density greater than 10^{15} particles per cu. cm. In the Los Alamos experiment the plasma is observed to force a passage through the input of the cusp under certain conditions of density and velocity and to remain in the cusp for the order of 40 microsec. Theoretically, if the magnetic field at the input to the cusp is made sufficiently strong, its pressure should lead to the exclusion of the plasma stream. This effect has so far not been observed in the cusp experiments in spite of extensive work.

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W. H. AILOR AND R. I. LINDBERG

CORROSION RESISTANCE OF ALUMINUM ALLOY 5083 IN MARINE ENVIRONMENTS

THE AUTHORS

William H. Ailor has been a member of the Chemical Metallurgy Section of the Metallurgical Research Division, Reynolds Metals Company, for six years. Educated in chemistry and chemical engineering at the University of Tampa, Florida, and North Carolina State College. He is Task Force chairman of the ASTM B-3/V1 1957 20-Year Test Program for the atmospheric corrosion of metals, secretary of ASTM Committee B-3 and is a member of the Electrochemical Society and the National Association of Corrosion Engineers. He formerly taught diesel engineering at North Carolina State College. He is a lieutenant commander in the Naval Reserve. During World War II he served as engineering officer afloat and later as commanding officer of USS PC 616 and USS Belet (APD-109). Recalled to service in the Korean conflict he served 17 months as executive officer of the USS Robinson (DD 562). He is in the active reserve and formerly was commanding officer of Surface Division 6-42 in Jacksonville, Fla. Currently he is training officer of MSTS Co. 5-1 in Richmond, Va.

R. I. Lindberg is a corrosion engineer in the engineering services department of Reynolds Metals Company. Author of several papers on corrosion in marine environments, Mr. Lindberg is a member of the National Association of Corrosion Engineers. Prior to joining Reynolds in 1959, Mr. Lindberg was director of corrosion research for A. M. Byers Co., Pittsbugh, Pa. Mr. Lindberg graduated from the University of Cincinnati in 1939 with B.S. and Ch.E. degrees. He took post graduate work at Purdue University receiving a Master of Science degree in 1942.

A NUMBER OF PAPERS discussing the corrosion resistance of aluminum have appeared. Many of these papers have dealt with long-time exposures and hence did not contain data on recently developed alloys. Some of the aluminum-magnesium alloys fall into this category and are of interest because of their excellent design characteristics. The following tables present information on the effects of exposing alloy 5083, both welded and bare, to marine environments for two years. Welding was done with 5356 wire using the inert gas consumable electrode process.

CHEMICAL COMPOSITION OF ALLOY 5083

Element	Alloy Tested	Specification (Max. unless shown as a range)
Si	0.13	0.40
Fe	0.22	0.40
Cu	0.02	0.10
Mn	0.38	0.30–1.0
Mg	4.42	4.0 -4.9
Zn	0.04	0.25
Cr	0.14	0.05–0.25
Ti	0.06	0.15
Others	—	0.15

TYPICAL MECHANICAL PROPERTIES OF ALLOY 5083

Tensile Strength .....46,000 psi
Yield Strength .....33,000 psi
Elongation in 2 inches ..... (15%)

Exposures were made at full submersion and at half-tide submersion at the Harbor Island facilities of the International Nickel Company. Atmospheric exposures were made at the 80-foot and 800-foot lots at the Inco Kure Beach Station.

As the data and photographs show, little corrosion occurs on alloy 5083 in marine exposures. No significant change in mechanical properties took place and maximum depth of corrosion was 3.0 mils (0.003″), occurring only on one sample.

Transverse section through pit in MD-21. 80 ft. lot. 250X. Maximum attack approximately 2.5 mils.

FIELD TEST RESULTS—ALLOY 5083

Atmospheric Exposure—Kure Beach 80′ Lot
4″x8″x0.25″ specimens

Condition	Time (Years)		Mechanical Properties T.S.	Y.S.	E(%)	Max. Pit Depth-mils
Bare	0	Control*	49,000	34,400	15.2	—
	1	Control	50,500	35,000	15.5	—
	1		50,600	34,800	15.1	—
	2	Control	50,200	34,100	15.2	—
	2		50,800	34,900	14.7	<0.5
Welded	1		44,200	—	—	—
	2	Control	44,800	—	12.2	—
	2		42,800	—	11.9	3.0

Atmospheric Exposure—Kure Beach 800′ Lot
4″x8″x0.25″ specimens

Bare	1		50,300	34,500	15.2	—
	2		50,600	34,400	13.9	<0.5
Welded	1		44,200	—	—	
	2		44,700	—	10.3	<0.5

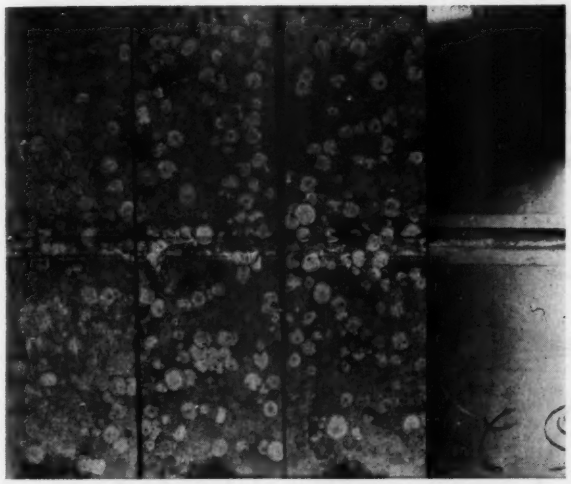
Half Tide Immersion—Harbor Island
4″x12″x0.25″ specimens

Bare	0	Control	49,000	34,400	15.2	—
	1	Control	50,500	35,000	15.5	—
	1		50,200	34,900	15.0	—
	2	Control	50,200	34,400	15.2	—
	2		50,700	34,300	14.2	<0.5
Welded	1	Control	44,900	—	—	—
	1		44,800	—	—	—
	2	Control	44,800	—	12.2	—
	2		43,500	—	10.0	2.0

Full Immersion—Harbor Island
4″x12″x0.25″ specimens

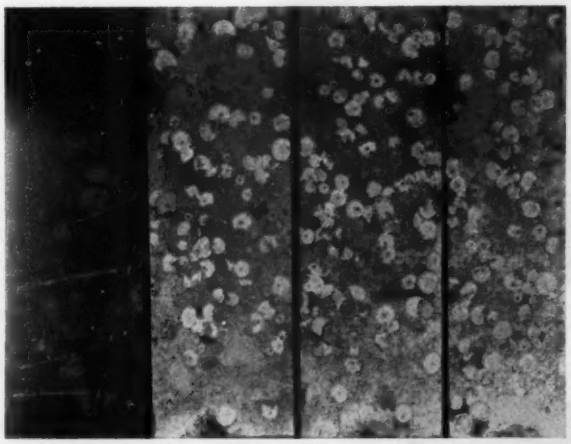
Bare	1	50,800	35,200	14.9	—
	2	50,400	34,400	15.1	1.5

* Control represents an unexposed sample.

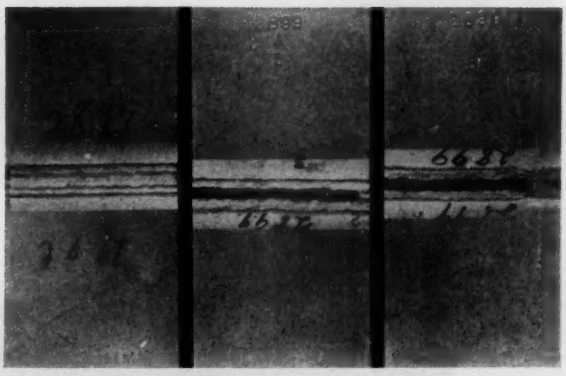


Welded 5083 (5356 Wire) exposed to half-tide immersion 2 years at Harbor Island, N. C. (Unexposed panel on right). White spots indicate areas from which barnacles were removed.

Two panels of 5083 on right were immersed 2 years in sea water at Harbor Island, N. C. All corrosion was superficial, even where barnacles were removed (white spots).



Bare 5083 (Unexposed Panel on Left) exposed to half tide immersion 2 years at Harbor Island (Unexposed Panel on Left).



Welded 5083 (5536 wire) exposed 2 years at Kure Beach, N. C. 80-foot lot marine atmosphere.

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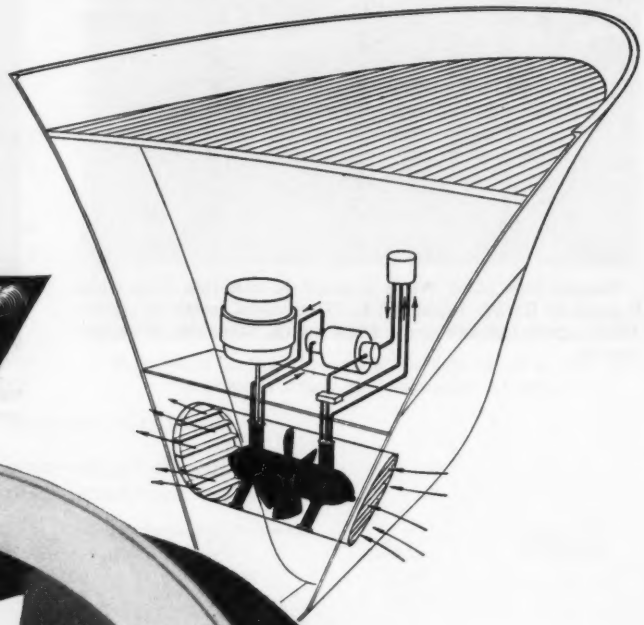

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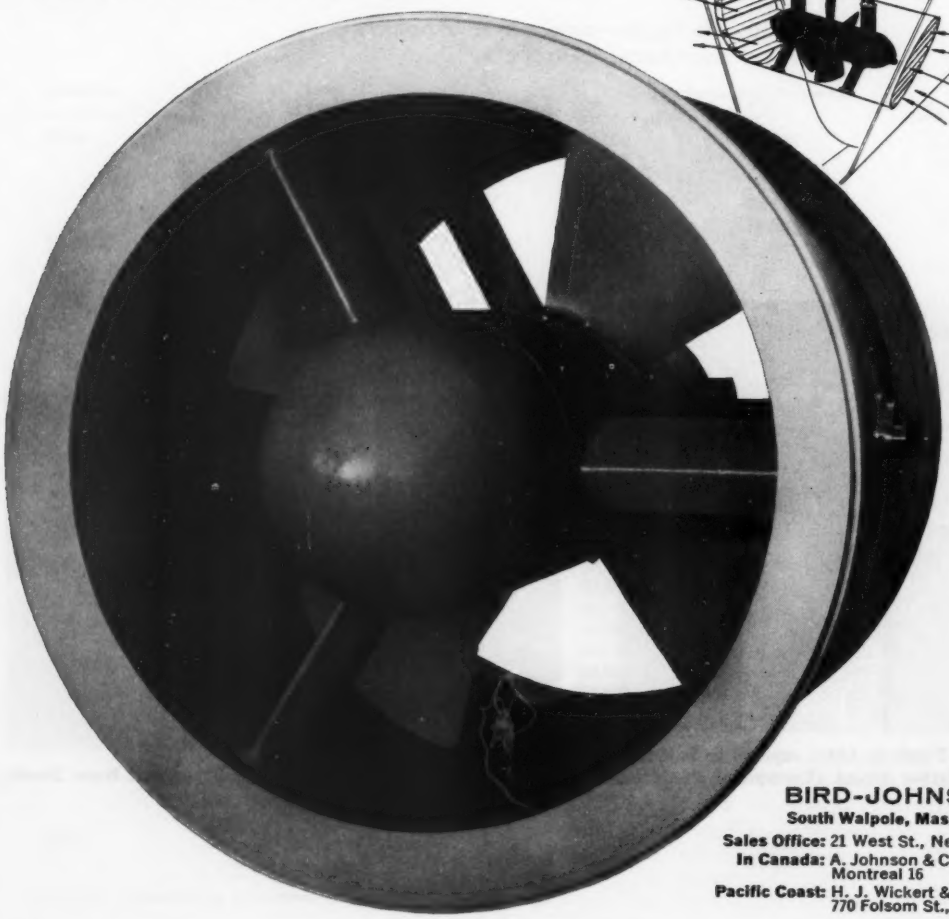

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"SHIPBUILDING AND SHIPPING RECORD"

THE FLUME STABILIZATION SYSTEM

ACKNOWLEDGMENT

This staff article appeared in the March 9, 1961 issue of "Shipbuilding and Shipping Record" from which it is reprinted.

SIMPLICITY in design is a valuable contribution to the success of any piece of engineering equipment, and although subsequent operational refinements may give the appearance of complexity this may be acceptable if the basic idea is straightforward.

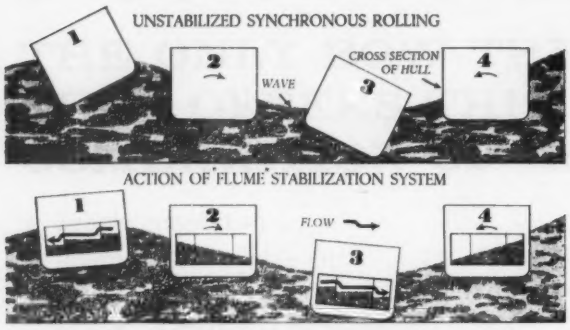
In the early part of this century Professor A. M. Robb wrote a paper describing the action of a ship in a seaway and the various aspects of stability and how it might be improved. A section of the paper was devoted to an explanation of the Frahm system of improving a ship's stability by making use of the movement of a quantity of water ballast through a series of athwartship tanks, which included wing and double bottom tanks.

The system was fitted to a number of ships but was not entirely successful, one of the disadvantages being the noise caused by the air venting between tanks.

A new development of the idea of using the rapid transfer of water between specially designed tanks to reduce the roll of a ship has been introduced by John J. McMullen Associates, Inc., New York, and it is stated that in terms of performance it represents a considerable advance on and is in no way comparable to the Frahm system.

The new anti-roll, passive tank layout, known as the Flume stabilization system, has only one moving part, the water, and consists of a pair of internal tanks, connected by a flume or duct.

In the Flume stabilization system, the weight of the ballast restricted to the high side is the source of the principal roll dampening effect. This principal stabilizing moment can be expressed as what in a heeled ship would be known as free surface effect, but in a rolling ship has been both amplified and changed in phase to become a pure damping moment. This control is effected by nozzles or restrictions, *i.e.*, rounding of the vertical corners where the flume and tanks are joined. This method of control causes the free surface moment vector of the partially filled tanks to lag 90°, thereby dampening the roll of the vessel, and at the same time, counteracting the tendency of the vessel to roll past the perpendicular of the wave slope. The system improves the roll stabilization relative to the perpindicular of the wave slope, thereby decreasing ap-



Diagrammatic illustration of the Flume principle.

parent roll to an even greater extent. This phase relation of roll is just as important as actual roll reduction because it is the roll away from apparent vertical which causes discomfort to passengers. The depth of water in the flume at the centerline plane of the ship remains the same throughout the transfer cycle and is dependent upon the total amount of water in the entire system rather than the movement of the vessel.

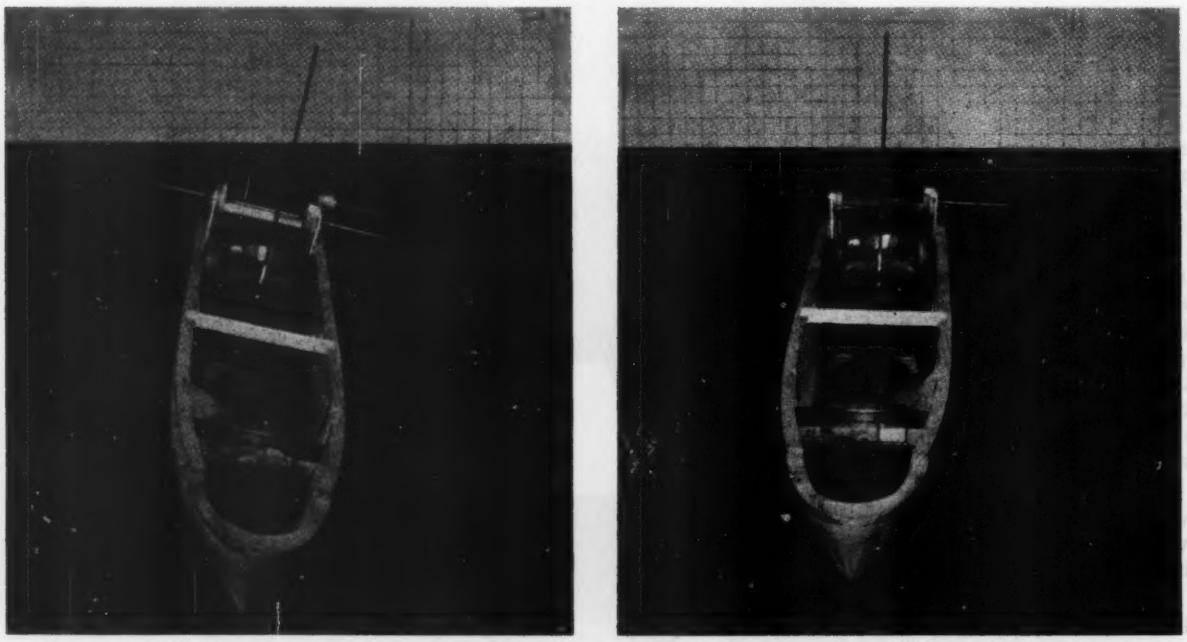
It is the flume nozzles that always keep the free surface moment at 90° away from the normal (static) position and acting about the axis of roll of the ship. They also prevent the vessel and the water from synchronizing as would occur if the system were too lightly damped, a potentially dangerous possibility in the earlier system.

In many respects the new arrangement could be likened to the partial flooding of two interconnected athwartship deep tanks. The interconnection makes it a modified U-tube type, as contrasted with the conventional or Frahm U-tube type. This interconnection provides the free surface, basic to operation of any passive tank system. The important differences are the addition, in the Flume system, of the control element, the nozzles, and the elimination of the air vent valves. The Flume method is effective at all speeds or hove-to.

The system has been installed in a variety of vessels ranging from the 27,000-ton passenger vessel *Matsonia* and the small research vessel *Vema* of the Lamont Geological Observatory of Columbia University to four missile tracking vessels of the United States Navy and several U.S. Navy icebreakers and oceanographic research vessels.

NO COMPLICATED CONTROLS

The Flume stabilization system has no mechanical parts or complicated control devices, so that no attention or maintenance is necessary and malfunctioning is impossible. There are not any appurtenances external to the hull itself, which eliminates drydock maintenance and inspection. Bilge keels are unnecessary and the combined reduction of roll and the absence of bilge keels results in a hull and wave resistance reduction of about 7 per cent. Operating expense is very small and, even more important, the system is in constant operation. Adjust-



Left: A 1/108th scale model rolling in regular swells (19 sec. period, 1.7° surface wave slope) at synchronous condition; the Flume system is inoperative. Right: Same model under same conditions but with Flume in operation. A 14° roll was dampened to 3°.

ments in the water level in the tanks are all that could be necessary to correct for variations in metacentric heights in cargo vessels. No special training or complicated calculations are needed. In the *Matsonia,* the water level is fixed and checked whenever convenient; no more than once per voyage. Visible through sight ports and illuminated by watertight lights are permanent height markings on the inside bulkhead for checking water levels.

In terms of capital investment, the Flume stabilization system costs less than 25 per cent of other mechanical methods of stabilization in the case of conversions, and even less when incorporated in new designs. Stabilization has to date been restricted by high initial and operating costs to passenger vessels, where the competitive demands of service made stabilization mandatory. On the other hand, the low initial cost and absence of operating expense of the Flume stabilization system permits its use not only in passenger vessels, but also in cargo ships of all types. Needless to say, where more than one vessel of a class is to be built, as is the case for most new construction, engineering costs per vessel are substantially reduced.

The amount of water ballast required in the Flume system varies between ships and is insignificant as a loss factor of either deadweight or cubic capacity in comparison to the total displacement of the ship. In the case of motor vessels, reserve diesel oil can be used in the system instead of water. On board the *Matsonia,* the 85 tons of water in the Flume amounts to less than ⅓ of 1 per cent of the total displacement. When use of the full deadweight capacity is more desirable than stabilization, such as in a fully loaded tanker, the flume ballast can be dumped or better yet, liquid cargoes can be carried in the Flume system.

Owners of some passenger vessels find the Flume system the only method they can consider because of 'midship space limitations, since this system can be located in any but the extreme areas of the vessel. In the *Vema,* the Flume system is located atop the pilot house! The *Matsonia* installation is located in the forward 18 feet of the lower No. 4 cargo hold, because of ease of installation. Since the space was not used for cargo, there was no loss to the ship.

Many of the less immediate returns on the investment in the Flume system will take time to evidence themselves.

The Flume stabilization system has been approved for installation by the U.S. Coast Guard and the American Bureau of Shipping.

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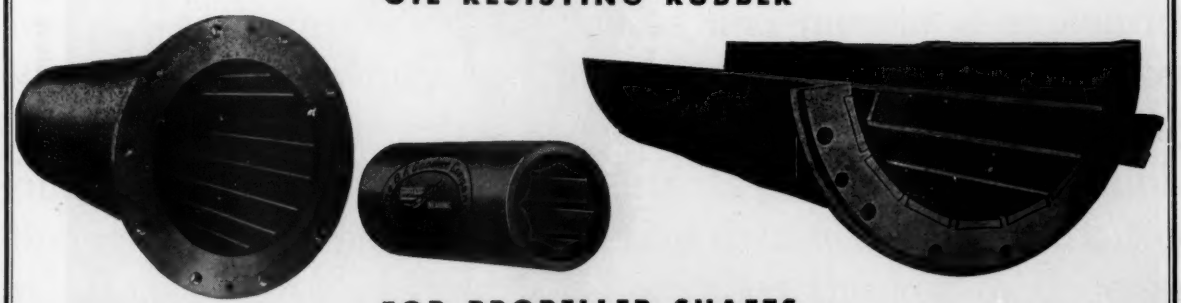

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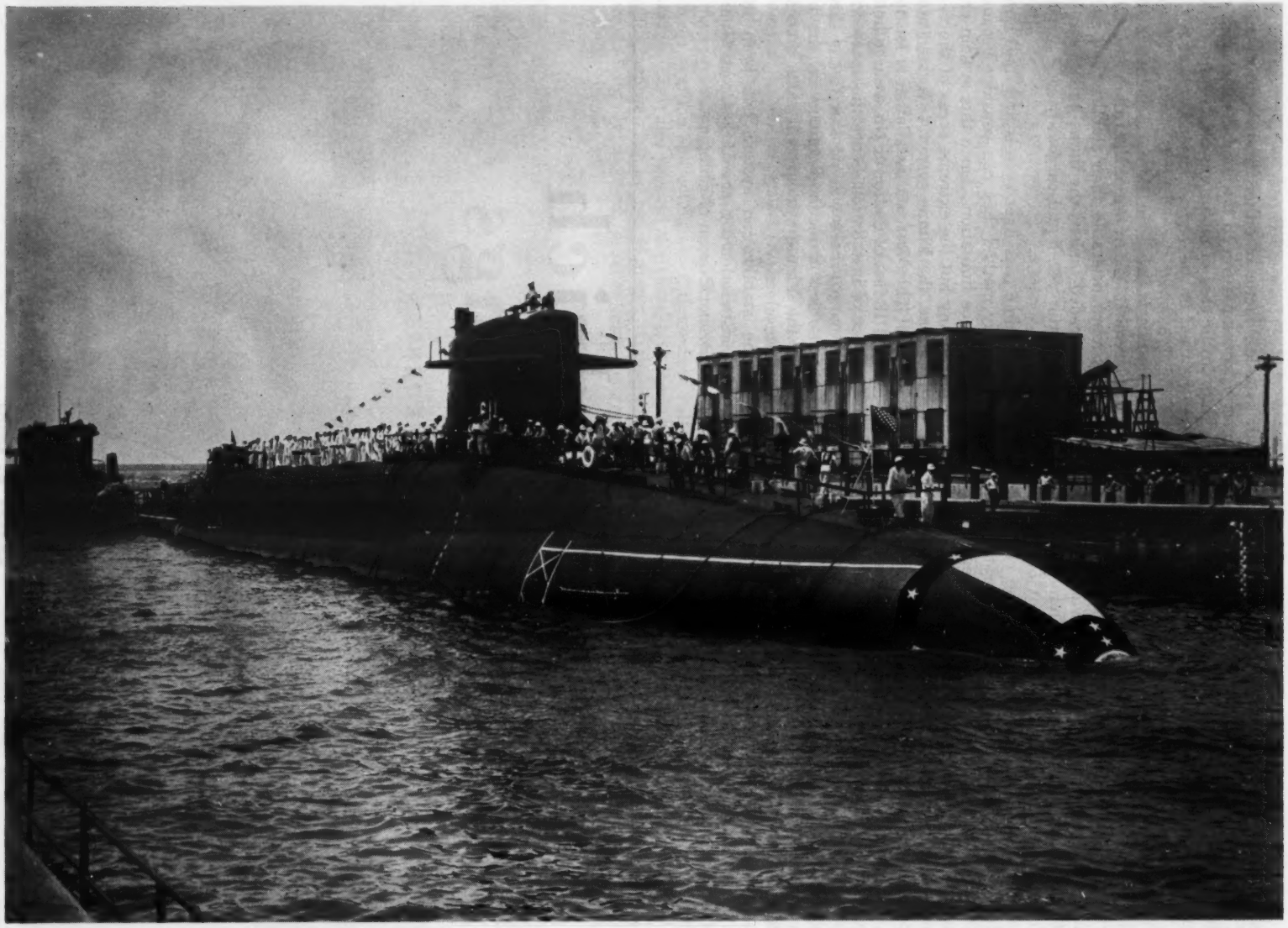
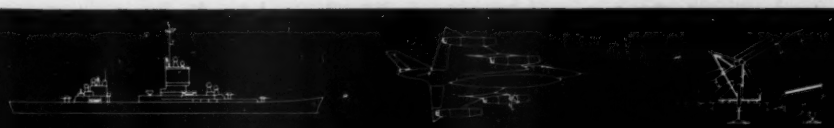


Fleet Ballistic Missile Submarine *John Marshall* SSB (N) 611, launched July 15, 1961.
Build by Newport News Shipbuilding & Dry Dock Co.

In Navy's Polaris missile system—a major contribution to free world defense—it's a matter of "as the submarine goes, so goes the missile."

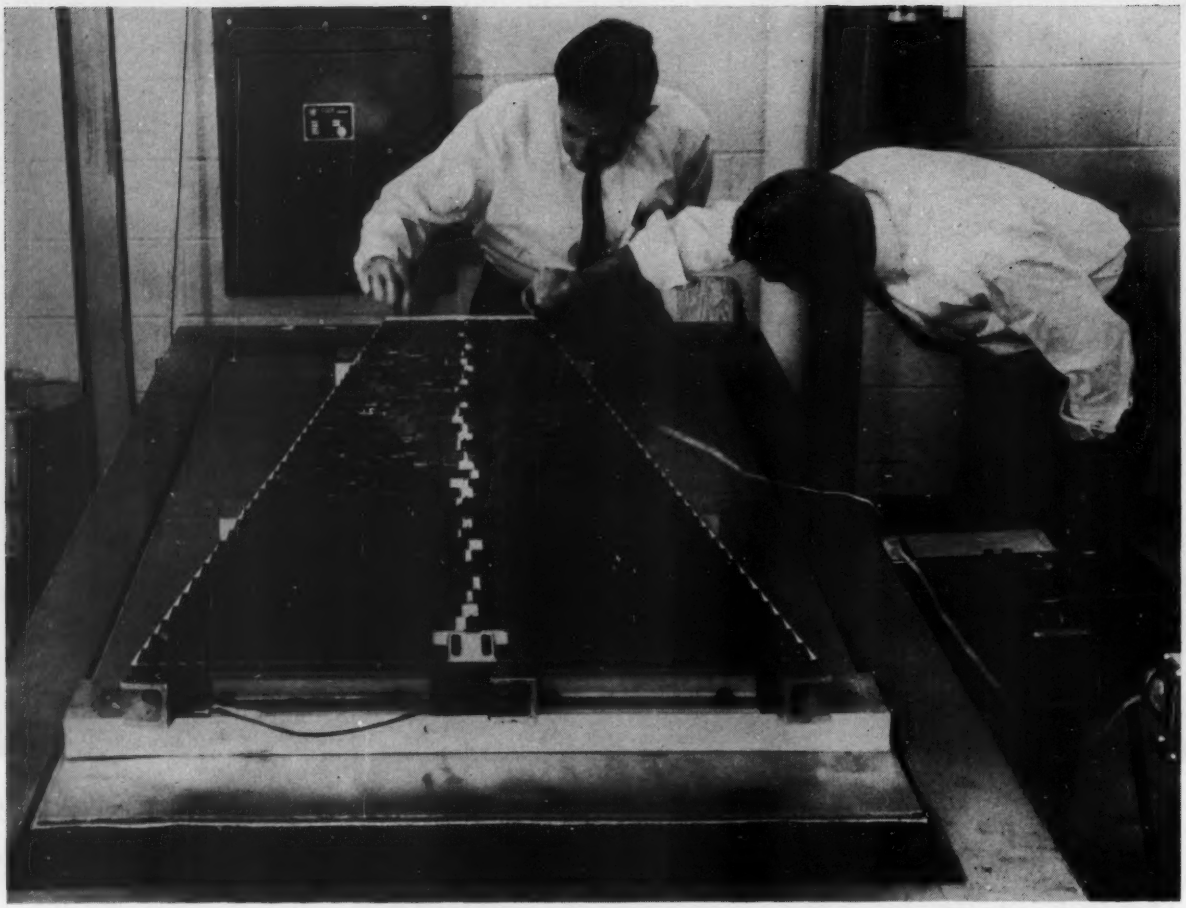
Navigation systems manager for this deterrent weapon, Sperry has evolved a system which provides the navigational accuracies required over the weeks and months a submarine is submerged. An inertial guidance system, double checked by a complex of instruments and master computer, not only guides the submarine and pinpoints its position, but telegraphs directly into the missile the exacting data needed to start it on its way. Thus has navigation been called the key to undersea firings: one degree error in the sub's heading means a 20-mile miss for the missile. General offices: Great Neck, N. Y.



Other Sperry contributions to deterrent weapons: precision acquisition, tracking and guidance radars for Navy's Terrier and Talos missiles; bomb-nav system for USAF's B-58; Army's Sergeant missile system; Sperry's Polaris submarine navigation simulator (main illustration)

SPERRY

Build by Newport News Shipbuilding & Dry Dock Co.



Solar power converter panel built for the JPL Ranger RA-1 Space Probe is given final check-out in laboratory. Although the panel will provide about 90 watts of continuous electrical power and contains 4340 individual silicon solar cells, it weighs only 19 lbs. Panel was built by the Semiconductor Division of Hoffman Electronics Corporation.

OVERVIEW OF SCIENCE PROBLEMS

In addition to the many technical enigmas demanding solution, industrial scientists are faced with four pressing non-technical challenges that must be met, according to Admiral Selden B. Spangler, Director of Research for The Garrett Corporation. Addressing the American Management Association at a recent meeting, Admiral Spangler pinpointed the problems of the increasing tendency to transfer basic research from universities to government and other special groups, lack of sufficient support for industrial applied research, lack of true devotion to the spirit of scientific investigation among technical graduates, arbitrary and unrealistic regulations regarding the free exchange of information among scientists. He warned also of the overpopularization of science, which, he said, "must inevitably lead to dilution of the true scientist's efforts through the actions of the hangers-on and the bandwagon riders." It is as wrong, he pointed out, to establish a scientific aristocracy as to brand scientists as 'eggheads' and impractical theorists; what is essential to the future is recognition of the value of all science and support of basic and applied research even in fields which are neither popular nor thoroughly understood.

—RESEARCH/DEVELOPMENT MAGAZINE

"BUSHIPS JOURNAL"

HISTORICAL DEVELOPMENTS OF THE MODERN SUBMARINE

ACKNOWLEDGMENT

This article, by E. M. Herrman of the U. S. Naval Engineering Experiment Station, Annapolis, Maryland, appeared in the March and April 1961 issues of "BUSHIPS JOURNAL" from which it is reprinted.

"THE WATERS OF THE bay were dark except where patches of misty moonlight flickered across the rippled surface, and the sky had just enough gray along the horizon so that the black silhouettes of ships stood etched against it. Conditions were ideal for undersea attack. One vessel, riding at anchor, loomed darker and more commanding than the others—the enemy flagship—and it was toward this that a midget American submarine stealthily made her way." [1]

The attack described above took place during the American Revolution, late in the evening of September 6, 1776, and is the first recorded instance of the use of an American submarine against an enemy ship. Before we describe the submarine involved in this attack, we shall discuss the development of submarines prior to that time. We shall then touch upon subsequent developments, including those of John Phillip Holland, and conclude by commenting on today's nuclear subs and the possible designs of the future.

In condensed papers such as this, it is impossible to treat historically *all* the submarine developments which have led to the present nuclear submarine. An attempt has been made, however, to present the highlights; and, at the same time, to detail information deemed interesting to readers of the *Journal*.

The information presented is largely a compilation of material found in the references listed at the end of this article.

EARLY EVIDENCE OF SUBMARINES

The earliest form of submarine was undoubtedly the human body. In 480 B.C., a Greek diver by the name of Syllics and his daughter, Hydna, played important roles in the defeat of the Persian fleet.

They cut the enemy's cables while the ships were riding out a storm off Mount Pelion. The Athenians are said to have used divers to clear the entrance to the harbor of Syracuse during the siege of that city. And in his operations against Tyre, Alexander the Great ordered divers to impede or destroy any submarine defenses the city might undertake to build. In none of the old records is there any direct reference to the use of submersible apparatus; however, a legend persists that Alexander the Great made a descent into the sea and observed a whale.

By 1400 a submersible in the shape of a cylinder with pointed bow and stern was produced and used for crossing rivers. In 1490 Leonardo da Vinci designed a rigid tube for use in exploring the mysteries of the sea; and in 1538 experiments with a diving bell were conducted at Toledo, Spain. Not until 1580 does any record appear of a craft designed to be navigated under water. In this year, William Bourne, a British naval officer, made designs of a completely enclosed boat which could be submerged and rowed under the surface. The boat consisted of a wooden framework covered with waterproof leather. It was to be submerged by reducing its volume as a result of contracting the sides through the use of hand vises. There is no evidence that an operating craft was ever produced from these designs.

However, Cornelius Van Drebel, a Dutch physician, carried out some interesting experiments on the Thames River in England. In 1620 he constructed two small boats of different sizes. These boats resembled those of Bourne in that their outer hull consisted of greased leather over wooden framework. The larger boat had 12 oars extending through the sides and sealed with tight-fitting leather flaps. These oars provided propulsion both on the surface and when submerged. This boat is credited with a voyage underwater from Westminister to Greenwich.

In the early years of the 18th century submersible craft seemed to abound. By 1727 no fewer than 14 types had been patented in England alone. In 1747 Symons introduced an ingenious device for submerging and surfacing a submarine: a number of goat skins attached to an aperture in the hull caused the vessel to submerge by allowing the skins to fill with water; and to bring it to the surface again, the water was forced out of the skins by hand and then the skins were tied.

In 1773 a ship's carpenter named Day constructed a wooden boat which was fitted with removable ballasts. Large stones with ring bolts were to be put aboard until the vessel sank and then let loose to allow the boat to rise again. Day sank this boat in 22 fathoms of water, which, apparently, was too much for the wooden structure; it collapsed. This seems to have been the first approach to the use of removable ballast.

We now come to the time when a submarine was

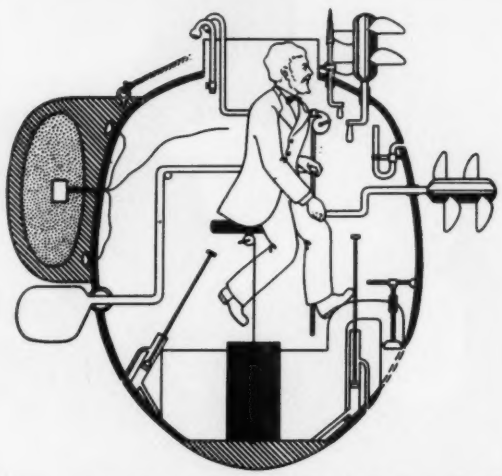


Figure 1. The *Turtle* of David Bushnell, from a drawing by Lt. F. M. Barber, USN, 1875.

first used as an offensive weapon in naval warfare. The first submarine used as an offensive weapon in naval warfare was invented by David Bushnell (figure 1) and was built in 1776. This was a one-man egg-shaped affair with two hand-cranked propellers: one for horizontal movement and one for vertical. Armament was a 150-lb. charge of gunpowder, which was attached to the target ship by a wood-screw-auger affair and exploded by a 30-minute timing device. (Thirty minutes was presumably long enough for the attacker to leave the area.) Air was supplied to the operator by snorkel tubes while surfaced, and enough air was trapped in the hull to keep the operator supplied while submerged for the attack. Maximum pressure was that corresponding to a depth of 18 to 20 feet; greater pressure than this caused the wooden seams to start oozing tar. Submerging was accomplished by flooding two tanks; surfacing was accomplished by pumping the tanks empty with two hand pumps. Several attacks were made by this submarine on the British ships off Long Island. The first attack, on 6 September 1776, failed because the British ship *Eagle* was sheathed on the bottom with copper as protection against barnacles, and the auger would not penetrate the copper. Although this sortie was abortive, it has been reported that General Washington complimented Dr. Bushnell on having so nearly succeeded in his attempt to sink the *Eagle*. A notable characteristic of Bushnell's submarine was its great inherent longitudinal stability. The stability described was the tendency of the submarine to remain right side up and not to dive to the bottom, unless the hull was punctured or flooded at one end.

Fulton's *Nautilus* (Figure 2) was started in May 1800 and completed in 1801 in France. It was probably the only submarine ever built which used sails as a means of surface propulsion. Submergence was

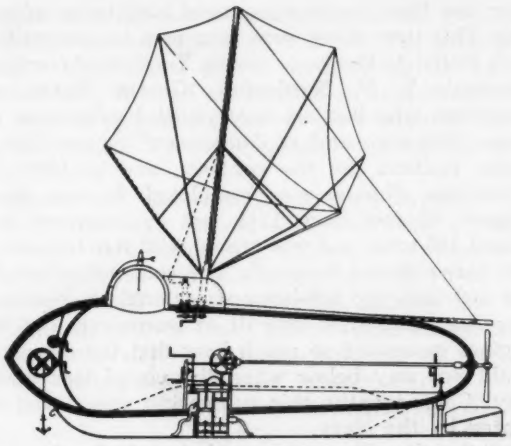


Figure 2. Fulton's *Nautilus*, about 1800.

accomplished by flooding ballast tanks; rising was accomplished by blowing the tanks with compressed air. Propulsion submerged was by hand. *Nautilus* made its first practice dive at Brest. Napoleon had contributed funds for the construction of the submarine, but was discouraged by its limitations. Among other shortcomings, it was too slow: top speed was approximately two knots with two crewmen cranking furiously. Fulton strengthened his boat so it could operate when submerged to 20 feet and could go to almost 30 feet. He added more compressed air tanks so he could stay down 1½ hours. After a demonstration in which *Nautilus* towed an explosive charge, at the end of a 200-yard cable, under an old schooner and blew it up Napoleon still was not convinced of the submarine's potentialities. He wanted one that would be effective against seaworthy ships while they were under way and one that could openly attack the British. Since Fulton could not meet these requirements, he offered to operate as a privateer, charging several thousand francs for every ship sunk. When a French admiral called submarine warfare ungallant, however, Fulton left the French and went to the British.

The British furnished funds for building a submarine and in the fall of 1805 Fulton blew up an old brig. The British Admiralty was so aghast at this unorthodox means of warfare that Fulton was offered a large sum of money to halt further development of his submarine. Fulton sailed for the United States. In America Fulton again demonstrated his destructive technique, but Navy officials wanted him to try a "dummy" run against a ship of the line. The crew of this ship, the sloop *Argus*, heard of the plan and placed nets around the ship, and caused Fulton's attack to fail. As a result of this failure, Fulton quit tinkering with submarines and invented the steamboat.

Between 1805 and 1863 little submarine work was done. However, in 1850 a Bavarian named Bauer built a submarine. It was controlled by shifting a weight forward to dive and aft to rise. It was a flat-sided, flat-decked vessel with comparatively thin plating and was entirely unsuited to the pressure of the water at considerable depth. It collapsed in the harbor of Kiel during one of its trial trips. This vessel remained partly buried in the mud into which it had sunk until 1887, when it was located during the deepening of Kiel harbor. It was taken to Berlin, where it is now kept in the Museum of Oceanography as Germany's first submarine.

During the American Civil War the South converted several gunboats into low armored rams, each of which carried a gunpowder charge on a spar at the bow. A partially successful attack was made by one of these boats against USS *New Ironsides* while she was lying off Morris Island in Charleston Harbor. (At this time three submarines, called "Divers" by Union people, were being built from old boilers designed by Captain McClintock of the Confedereacy.) Two attempts to use these rams proved futile. In one, the ram was scuttled to prevent capture; and in the other, the ram was sunk during a squall while it was attacking a Federal fleet in Mobile Bay.

Another ram, *Hunley* (Figure 3), named for the man who donated money to build it, was a "jinx" from the start. During the first trial run in the summer of 1863, her nose stuck in the mud and all seven men aboard suffocated. She was salvaged, and manned by a new crew under Captain Payne. During a trial run *Hunley* was swamped by a passing steamer and six of her seven-man crew drowned. A week later she was swamped again, but this time Payne and two others escaped. Payne finally relinquished his command. The submarine was taken out for the fourth time, however, and two more crew members were drowned. *Hunley* was raised, and made a practice run on the Confederate ship *Indian Chief*. She became fouled on an anchor chain and sank in 9 fathoms. Salvaged by Lt. Dixon, *Hunley* made one last sortie against USS *Housatonic*. This, too, was unsuccessful. Although *Housa-*

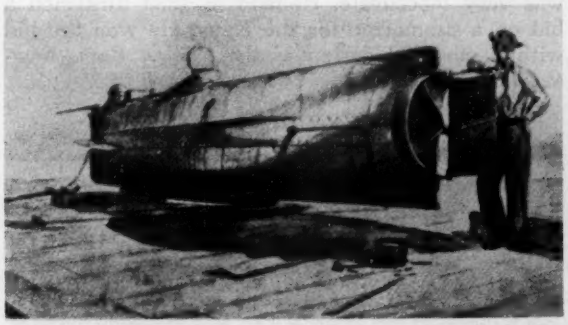


Figure 3. Sketch of the Confederate submarine *Hunley*, made subsequent to her recovery years after the Civil War. (Drawing by R. S. Skerrett.)

tonic was sunk, she rolled over on *Hunley* and took her down. The use of *Hunley* demonstrated perseverance in attempting to make submersible vessels practicable; however, no description exists of an attempt to analyze her problems and correct her instability.

During 1863 and 1864 Messrs. Bourgois and Brun built for the French Navy the largest and most completely equipped submarine that was produced during the 19th century. This was *Le Plongeur*, a vessel about 140 feet long, 10 feet in depth, and with a beam of 20 feet; displacement was over 400 tons. She was propelled by compressed-air engines which produced a total of 80 horsepower. The compressed air was carried in tanks at a pressure of 180 psi and the capacity of the tanks was approximately 4950 cubic feet.

WORK OF JOHN HOLLAND

"The submarine is an American invention, . . . the genius of an ardent Irish-American patriot, John P. Holland." [4] John Holland came to America in 1873, and reportedly began his work on submarines as a result of an accident. On one of his visits to the New York Public Library he slipped and broke his leg; this injury confined him to his room for a period of three months. Holland used this time to work on plans for a submarine, the building of which was supported by a group of Irish rebels called the Fenians (Figure 4). The submarine was powered by a Brayton gasoline engine, invented in 1874. Holland's first submarine was not very successful; it would submerge and surface, but it would not run submerged.

Holland designed another submarine which was 30 feet long and had an air gun forward. The Fenians contributed $13,000 toward the construction of this vessel which was built in the Delameter Machine Shop in New York. When completed, the submarine could travel 6 knots submerged and 8 or 9 knots on the surface.

In 1886 Holland built a submarine for the Army, in cooperation with a Lt. Zalinski. It was, more or less, an underwater gun-carriage. It never worked, it made only 3 knots, it was unstable, and the gun was very inaccurate. Holland in 1889 furnished a bid on a submarine for the Navy. He won the bid with a modified design of the *Fenian Ram*; however, the contract was never awarded. Five years later the Navy again announced bids for a submarine. This time there were four men in competition with Holland: Professor Josiah Tuck, an American; Sweden's T. V. Nordenfelt; George Baker, an American who had an experimental submarine on Lake Michigan; and an "unknown" named Simon Lake. Holland got the contract, and in 1897 the submarine *Plunger* was completed. It was cigar-shaped; 85 feet long; 11½ feet in diameter; displaced 186 tons; and was armed with five torpedoes, two forward and three aft. The contract called for one 300- and two 600-horsepower engines. Not only was the submarine difficult to maneuver, but the engines generated so much heat that the operators could not stay below when the vessel was under way. Consequently, this submarine was never accepted by the Navy.

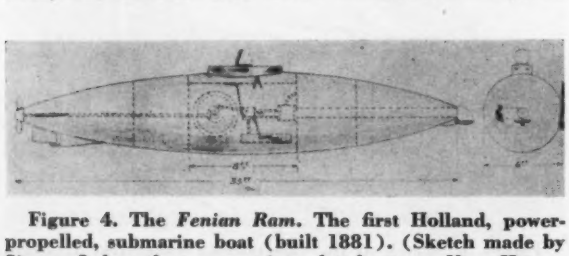


Figure 4. The *Fenian Ram*. The first Holland, power-propelled, submarine boat (built 1881). (Sketch made by Simon Lake after measuring the boat at New Haven, Connecticut, in 1915.)

While the *Plunger* was being built, Holland started work on a submarine with his own money, plus an additional sum of $25,000 from a New York backer. This submarine was completed, but a careless worker left a valve open and the boat sank at the pier. The boat was raised and the electrical system was dried out by a representative of the Electro-Dynamic Company of Philadelphia, Frank Cable, who became the submarine's first civilian skipper [4]. This submarine was named *Holland* (Figure 5) and was 53 feet long and 10 feet in diameter. When in diving trim, she had 100-lb. reserve buoyancy. She carried 12 tons of main-ballast water, plus ½ ton in each trim tank; and sixteen removable 50-lb. lead weights on deck. Diving was controlled manually from the conning tower by a rack-and-pinion arrangement.

Holland had a ⅝-inch-thick hull. It could and did operate at a depth of 60 feet. Open battery cells prohibited diving or surfacing at an angle greater than 15°. A 150-hp electric motor produced a maximum submerged speed of 9 knots. A 120-hp, 2-cycle, 15-

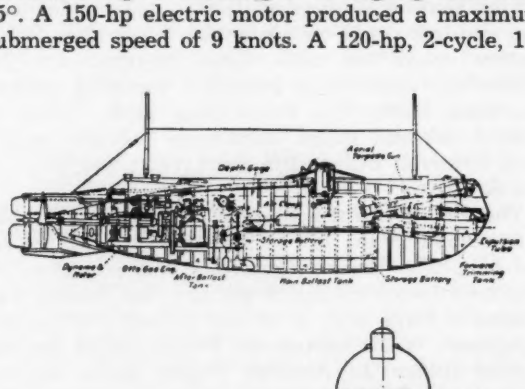


Figure 5. The *Holland*. This vessel, while holding to the same general principles of construction and method of control as used in *Plunger*, was much better proportioned and had a much better distribution of weight. It was her performance that led the House Naval Committee, in 1900, to authorize construction of additional submarines of the *Holland* type.

inch-stroke gas engine, operating at 400 rpm, produced a surface speed of 8 knots. The submarine had a four-bladed propeller, mounted forward of the rudder. A small compressed-air engine was used to operate the rudder and to blow water from the ballast tanks. The submarine had a surface draft of 7 feet.

Seven men made up the crew. A dynamite gun aft (it was later moved forward) fired a 6-inch-diameter by 8-inches-long cylinder of dynamite by compressed air and gunpowder for 900 yards. The submarine had poor longitudinal stability and was difficult to steer. A leaky exhaust pipe nearly killed the crew. (Later white mice were carried to warn of impending danger of air contamination.)

On July 4, 1898 Mr. Isaac Rice, of the Electric Storage Battery Company (which furnished *Holland's* and *Plunger's* batteries), bought into the Holland Company and the Electric Boat Company was formed. In March 1900 *Holland* passed Navy acceptance trials. On April 18, 1900 the Navy bought *Holland* for $100,000. Lt. C. H. Caldwell, then an aide to Admiral Dewey, requested and received command of *Holland*. In 1901 John Holland built a submarine 63 feet long, of 120 tons displacement, named *Fulton*. This submarine was eventually sold to Russia.

Japan ordered and obtained five submarines, which were assembled in Yokohama. These boats were built in sections at the Fore River Works, Quincy, Massachusetts, with approximately the same lines as *Fulton*. The hulls were designed to withstand a depth of 125 feet. The dimensions were:

Length	58 feet 9 inches
Maximum depth	11 feet 9 inches
Surface speed	9 knots
Submerged speed	8 knots

The hulls had bilge keels 9 inches wide extending for a length of 24 feet 6 inches; and were made of ⅜-inch plate. There is no report that these boats played any important part in the Russo-Japanese War of 1904-1905. By 1903, the U.S. Navy had ordered six new boats ("A" Class), and *Plunger* was also rebuilt and commissioned. *Adder* (A-2) was the first of the new "A" boats and was something of a show piece; at times it made as many as 90 dives a day, with officials aboard. All of these "A" boats were used for training personnel and for developing submarine warfare techniques. All but the old *Plunger* were eventually used as target vessels in 1922. *Holland* was decommissioned in 1906 and was given to the N. Y. Central Park Authority, which had it on display for a while and scrapped it in 1930.

SUBMARINES OF SIMON LAKE

Although Lake did not obtain the first contract to build a submarine for the Navy, he was not discouraged. In 1894 he built a boat that was 14 feet long and had a beam of 4½ feet. This boat was constructed of two layers of yellow pine. The inside layer was lined with felt; the outside layer was well caulked, and covered with tar. The boat was called *Argonaut, Jr.* (Figure 6). Two large driving wheels were attached to the hull; these wheels were connected through chain gearing to sprocket wheels inside the boat. By manually turning the inside wheels the vessel could be propelled along the river bottom. Lake was particularly interested in using his boats for salvage operations. By closing all openings in a compartment and precharging it with air to balance the sea pressure, a diving hatch could be opened from which a diver could leave or enter at will. A boat constructed by Lake in 1902 was similar to that shown in Figures 7 and 8.

SUBMARINES OF OTHER COUNTRIES

As a result of the success of Holland's work, Britain and Russia became more interested in submarine development. Holland built a submarine for

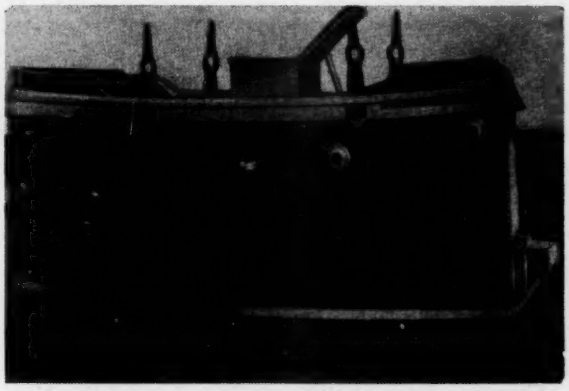


Figure 6. The *Argonaut, Jr.*

Figure 7. Submarine with cushioned bottom wheels. The drawing illustrates how such a vessel will surmount a steep declivity, while a boat of the diving type (D) will either "bury her nose" or strike with sufficient force to disarrange her machinery.

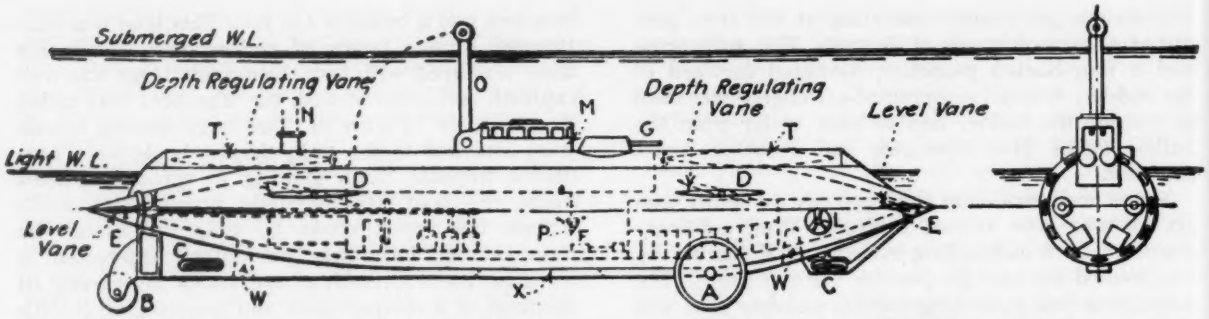


Figure 8. Lake design, submitted to the U. S. Navy Department in 1893. Novel features included: (A) wheels for running on the bottom; (B) a rudder which functioned as a steering wheel when the vessel was navigating on the bottom; (C-C) propellers for holding the vessel to depth when not underway; (D-D) depth-regulating vanes for causing the vessel to change depth while underway; (E-E) horizontal rudders, or "leveling vanes," designed to automatically hold the vessel on a level keel when underway; (F) a weight, controlled automatically by a pendulum; (P) mechanism to correct trim; (G) gun, in watertight revolving turret; (L) propeller in tube for swinging vessel at rest to facilitate "pointing" her torpedoes; (M) conning tower; (N) telescoping smokestack; (O) observing instrument, arranged to turn down on deck when underway; (T-T) torpedo tubes, two firing forward and two aft; (W-W) anchoring weights to hold the vessel at rest at any desired depth; (X) an "emergency keel," released automatically when the vessel reached an unsafe depth.

the British and tested it succesfully in the Irish Sea. Five submarines which included modifications determined from the tests of the first *Holland* were built for the British. Built about 1903, these submarines had the following characteristics: length 63 feet 4 inches; beam 11 feet 9 inches; height 12 feet 1 inch; displacement 120 tons. The hull was constructed of circular transverse sections with plating laid in ten strakes 7/20-inch thick forward and aft and 8/20-inch thick amidships. Frames were 3½ feet by 3 inches by 7.8 lb. steel angles and were spaced 18 inches apart. Stringers were 2½ inches by 2 inches by 5.9 lb. steel angles. Bulkheads were 17½ lb.-per-sq.-ft. steel plate; and deck beams were 3 inches by 3 inches by 6.1 lb. angles covered by 17½-lb. and 15-lb. plates. Design depth was 100 feet.

The first Russian submarine, built in 1879, had a steel hull 14 feet long and a maximum diameter of 5 feet.

Germany began building submarines of the Nordenfelt type in 1890 at Kiel and Danzig. The Nordenfelt propulsion system consisted of steam-driven, 100-hp, reciprocating engines of 100 psig, with cranks placed at an angle of 90°. The boiler was a return-tube type with one furnace. When the vessel was submerged power was obtained from steam derived from 8 tons of stored hot water. Sufficient power was obtained to give an underwater range of 15 miles at 4 knots.

The dimensions of these submarines were:

Length	34.85 meters
Diameter	3.65 meters
Surface displacement	212 tons
Speed	11 knots
Submerged speed	4.5 knots

In August 1901 the *U-1* was built by the Germans. This submarine had the following characteristics:

Length	101 feet 3 inches
Diameter	10 feet 1½ inches
Displacement	240 tons
Speed (surface)	11 knots
Speed (submerged)	9 knots

Armament consisted of one bow tube and three torpedoes. During 1906-07 Germany increased her submarine fleet to 144.

Since Russia and Japan began to show considerable interest in submarines produced in his newly-formed submarine-building yard, Simon Lake became more active in the construction of undersea craft and began to compete with Holland. Lake built submarines for Russia in 1906. These boats could remain submerged for a mile at a depth of 126 feet. Lake defined a modern submarine: [5] "A modern submarine may be described as a complex machinery mechanism capable of being navigated on the surface of the water just as is any boat but with the added faculty of disappearing at will beneath the surface and of being operated beneath the surface in any desired direction at any desired depth." This definition applies in general, today; however, we are limited in depth of operation with our present-day submarines. Lake had the idea that his submarines could operate on the bottom of all the seas. Figure 7 shows his idea for preventing a submarine from burying its bow in the bottom.

The United States Navy began to consider the submarine a potential military weapon around 1904.

USE OF DIESEL ENGINES

The development of the diesel engine added impetus to the development of the submarine. The first serious consideration of the use of diesel engines in submarines appears to have been in 1911, when the Electric Boat Company contracted with the Bethlehem Shipbuilding Company in Quincy, Massachusetts to build diesel engines, from Vickers

Figure 9. Submarine *Seal*—Lake-type U.S. This vessel was the first to be built that was provided with hull tubes and deck torpedo tubes. The latter could be trained and fired to either broadside while the vessel was submerged. In her acceptance trials the vessel descended to a depth of 265 feet. She broke the record for speed in the U.S. Navy.

drawings, for submarines. The U. S. Types E and F submarines had two 4-cylinder engines, each of which produced 400 bhp at 400 rpm. The engines operated on a 4-stroke cycle and had air injection of the fuel. By the end of 1917 Busch-Sulzer had about 70 engines either built or being built. All were air-injection types. The diesel engine continued through various stages of development until World War II, when General Motors 278A and Fairbanks-Morse 38D8-1/8 became the standard propulsion engines of U. S. Navy submarines. After World War II, improvements in weight reduction were developed, but it was not until the advent of nuclear power that a true submersible was produced.

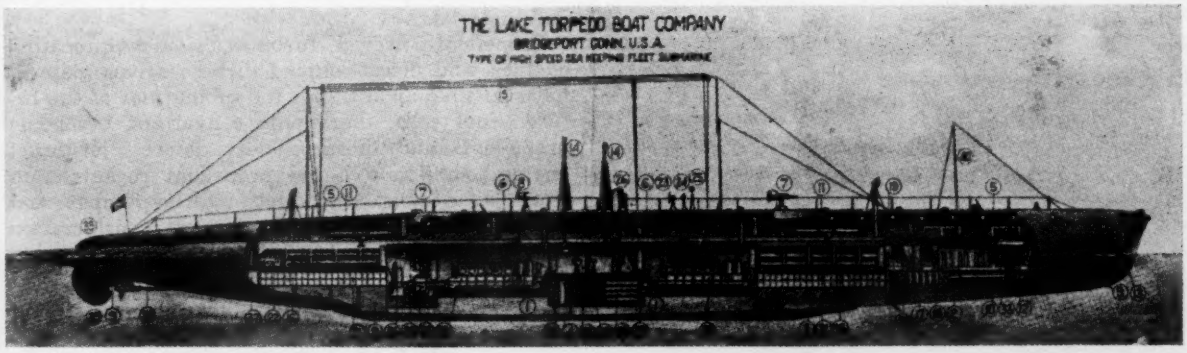


Figure 10. A submarine cruiser, or fleet submarine (Lake type). Indicated by numbers are: 1, main ballast tanks; 2, fuel tanks; 3, keel; 4, safety drop keel; 5, habitable superstructure; 6, escape and safety chambers; 7, disappearing anti-aircraft guns; 8, rapid-fire gun; 9, torpedo tubes; 10, torpedoes; 11, twin deck torpedo tubes; 12, torpedo firing tank; 13, anchor; 14, periscopes; 15, wireless; 16, crew's quarters; 17, officers' quarters; 18, warhead stowage; 19, torpedo hatch; 20, diving chamber; 21, electric storage battery; 22, galley; 23, steering gear; 24, binnacle; 25, searchlight; 26, conning tower; 27, diving station; 28, control tank; 29, compressed-air flasks; 30, forward engine room; 31, after engine room; 32, central control compartment; 33, torpedo room; 34, electric motor room; 35, switchboard; 36, ballast pump; 37, auxiliary machinery room; 38, hydroplane; 39, vertical rudders; 40, signal masts.

MODERN SUBMARINES

The Navy's first nuclear submarine was the pioneering, one-of-a-kind *Nautilus*, which was commissioned in 1955. Next came *Sea Wolf* which, like its predecessor, was a special developmental model designed for attacking surface craft. In December 1957 the Navy commissioned the first production atomic submarine, *Skate*, which also was an attack submarine. Since then, three more submarines of the *Skate* class have been commissioned. As the next step in nulcear-submarine planning, six attack subs of radically new design and one giant radar-picket submarine were ordered. The new attack class is the *Skipjack*, whose shark-shaped hull and single propeller produce high, sustained, underwater speeds and improve the craft's maneuverability. The last of these six *Skipjack* subs will be commissioned by the middle of 1961.

The radar-picket *Triton*, which is a 5900-ton ship, is the largest submarine ever built. Commissioned in November 1959, she is fitted with high-powered radar and sonar. *Triton* will run ahead of high-speed Carrier Task Forces, and will detect and report on approaching enemy planes, submarines, and surface vessels. Also commissioned in 1959 was *Halibut*, the first nuclear submarine to carry a guided missile. *Halibut* can fire the air-breathing "Regulus" from the surface. Fourteen models of an advanced version of the *Skipjack* are also being built. All of these attack craft, known as the *Thresher* Class, are to be commissioned by 1963. The need for combating enemy submarines in their own environment led to the construction of the nuclear-powered killer-submarine, *Tullibee*, commissioned in late 1960. The *Polaris* ballistic missile will be in the Fleet Ballistic Missile (FBM) sub-

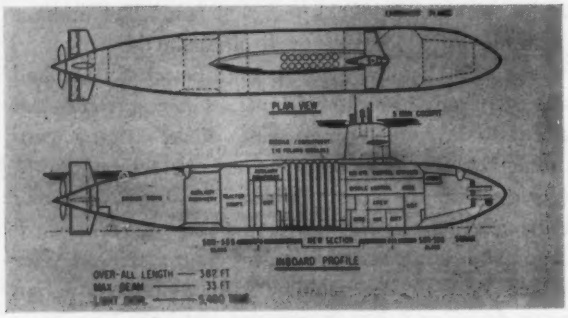


Figure 11. SSBN-598 Class.

marines which are designed expressly as missile carriers or launching platforms. The first of these submarines, *George Washington,* became operational in 1960. The next two FBM submarines became operational in the same year. Another 11 FBM submarines are on order.

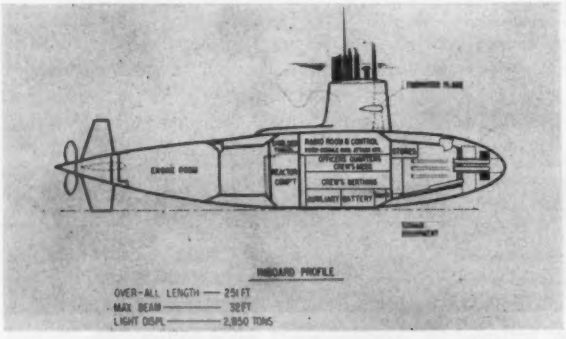


Figure 12. SSN-585, SSN-588-592 Class.

FUTURE DESIGNS

Submarines of the future may include small, reconnoitering types and giant bomber types. They may be armed with external missiles or provided with drone escorts. Eventually, they may even be backed up by unmanned-satellite stations or by ocean-bottom armories and fueling stations which would carry out pre-programmed missions upon command.

Future ships may be designed so that the entire power plant can be replaced as a unit, much as an engine in a plane. Designs may include a control capsule to permit the crew to bail out, if the submarine should get into trouble.

The trend in submarine performance is to dive ever deeper and faster. Whole new concepts of design will be required for this to be accomplished on an ever-increasing scale. Nuclear power has given the submarine endurance and mobility of magnitude beyond those of its snorkeling, diesel-powered predecessors. The solid-propellent ballistic missile with a nuclear warhead has given the submarine a destructiveness without precedent in naval warfare.

An understanding of all phases of hydrodynamic theory is basic to the solution of our present problems. To obtain more knowledge of ocean depths, we also need much more knowledge of structures and materials.

Propellent-powered turbines, slow-reciprocating engines, and direct-coupled turbine drives may all have possible applications for submarines of the future. Fuel cells, thermionic converters, cryogenic propellents and thermonuclear devices, jet-thrust engines, solid and liquid propellent rockets, ram jets, pulse jets, turbo-jets, turbo-pumps, and magneto-motive-pump jets all merit consideration in relation to the underwater craft of the years to come.

The development of the submarine, up to very recent years, occurred largely on the basis of cut-and-try methods. Today's designers, however, are using available hydrodynamic theory and information regarding statical and dynamical stability.

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NUCLEAR POWER FOR SPACE

Recent breakthroughs in several fronts on direct-energy conversion promise to make nuclear-space power packages practical in the next few years. Plasma diodes, thermionic and thermo-electric devices are proving out in experimental operation. They will become an integral part of small fission reactors to be used for power in advanced satellites.

—THE IRON AGE, May 25, 1961

"TECH ENGINEERING NEWS"

MOLECULAR ELECTRONICS

ACKNOWLEDGMENT

This article by Fred Philip Isaacs appeared in the February 1961 issue of "Tech Engineering News" from which it is reprinted.

THE GREATEST TREND in electronics today is miniaturization. The transistor has been a major step forward in this direction. But utilization of a new concept in solid state devices, molecular electronics, holds even greater promise. There are three major goals in the design of any electrical device. First, the apparatus must be dependable—"fail safe," if possible. Second, power losses due to heat evolution (I^2R losses) must be minimized. And third, the apparatus must be made as light and compact as possible.

How do molecular electronic, or "molectronic," components fulfill these requirements? First, as to dependability: one single molectronic "function block" can perform the same task as several conventional components. Thus, each function block is a subsystem (such as an amplifier, multivibrator, or a multiple switch) rather than a component (such as a diode or triode). Obviously, there are fewer connections and constituent parts. The subsystem is composed entirely of semi-conductor zones and interfaces, which are quite stable, barring physical damage to the unit. Hence, the function block is inherently more stable than conventional subsystems, by virtue of Murphy's Law. There are simply fewer things to go wrong.

Because of the compactness of this sort of design, and the lack of connections, one would naturally expect lower I^2R losses. There is simply less available resistance. But the most striking contrast between molectronic and conventional systems is that of size and weight. A cigarette pack sized transistor radio, excluding power supply and speaker, can be miniaturized, by means of molectronic techniques, to the size of a match head. Figure 1 compares three designs for a light telemetry subsystem, and demonstrates the simplification and miniaturization effected by both semiconductor and molectronic techniques.

What advances have made this type of system possible? Molectronics owes its very existence to the transistor. Without recent advances in semiconductor production techniques, it would be impossible to build function blocks, especially on anything like a commercial basis. The dendrite method of growing germanium crystals is of particularly great importance. Long ribbons of germanium, about an eighth of an inch wide and a few thousandths of an inch thick, are drawn from a bath of molten germanium. Each of these strips is a single crystal, and hence segments of it will be reasonably uniform in composition and molecular orientation. Any impurity added to the melt during growth will be confined to a single region of the crystal. The crystal is thus "self healing." As the surfaces of these dendrites are optical flats, semiconductor de-

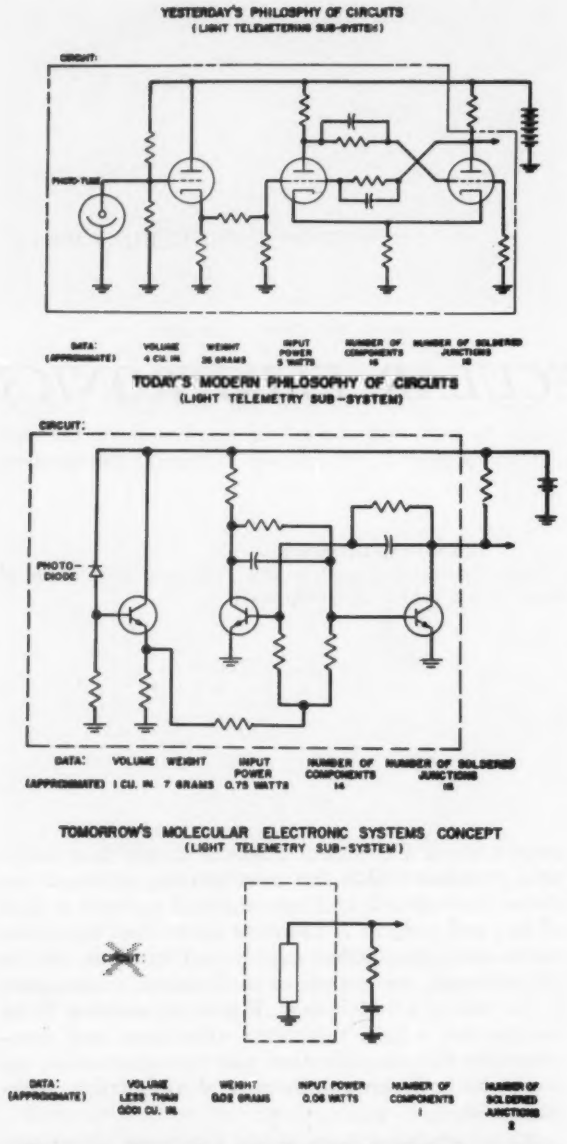
YESTERDAY'S PHILOSOPHY OF CIRCUITS
(LIGHT TELEMETERING SUB-SYSTEM)
CIRCUIT:
PHOTO-TUBE
DATA: (APPROXIMATE) VOLUME 4 CU. IN. WEIGHT 35 GRAMS INPUT POWER 5 WATTS NUMBER OF COMPONENTS 16 NUMBER OF SOLDERED JUNCTIONS 10
TODAY'S MODERN PHILOSOPHY OF CIRCUITS
(LIGHT TELEMETRY SUB-SYSTEM)
CIRCUIT:
PHOTO-DIODE
DATA: (APPROXIMATE) VOLUME 1 CU. IN. WEIGHT 7 GRAMS INPUT POWER 0.75 WATTS NUMBER OF COMPONENTS 14 NUMBER OF SOLDERED JUNCTIONS 18
TOMORROW'S MOLECULAR ELECTRONIC SYSTEMS CONCEPT
(LIGHT TELEMETRY SUB-SYSTEM)
CIRCUIT:
DATA: (APPROXIMATE) VOLUME LESS THAN 0.001 CU. IN. WEIGHT 0.05 GRAMS INPUT POWER 0.06 WATTS NUMBER OF COMPONENTS 1 NUMBER OF SOLDERED JUNCTIONS 2

Figure 1. Comparison of Yesterday's, Today's, and Tomorrow's Design of a Light Telemetry Subsystem.

vices can be assembled without any preliminary grinding or polishing. Using this type of crystal, it is literally possible to cut off the length of semiconductor needed to assemble any given device. Also, there are now methods of adding impurities to a growing crystal of germanium, and confining them only to desired regions. The combination of these two processes may well yield a method of producing function blocks on an assembly line basis.

A function block is designed in the following manner. An electrical engineer, knowing the use to which the unit is to be put, decides upon a "philosophy of design," and the materials to be used. This philosophy of design is simply the general arrangement of domains and interfaces within the function block. Then a topologist computes the most effective domain and interface shapes. Once this has been done, the required impurities are added to a germanium crystal of the correct size and shape, and a function block results.

The basic design philosophy underlying molectronics is quite intriguing. Each block consists of several molecular domains and the interfaces between them. Dr. Herwald, of Westinghouse, has drawn a particularly good analogy. In any given function block, there are ". . . a number of distinct operative domains, which can be regarded as molecular 'communities' having a common civic purpose, in that each domain will sustain a desired electronic occurrence. The domains border one another at boundaries called interfaces, which are like political frontiers in their ability to initiate phenomena different from those occurring inside the molecular domains". Figure 2 exemplifies this concept. This subsystem consists of a resistive domain, which converts electrical energy to heat, by means of I^2R losses. A second layer of a thermal insulator provides a steady heat flow to a thermoelectric domain. Thus, we have an extremely simple rectifier. Figure 3 represents a molectronic time delay circuit. The interface between the two resistive domains provides a capacitive effect. Thus, we have a molectronic capacitance in series with a resistance—an R-C, or time delay, circuit.

It has been stated that molectronic systems

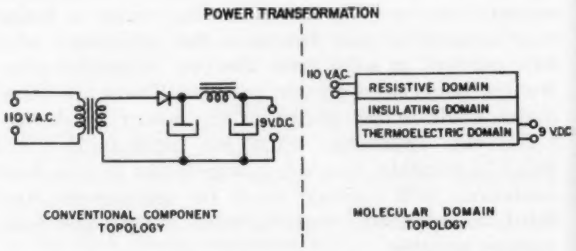

Figure 2. Power Transformation Subsystems—Molectronic vs. Conventional Viewpoint.

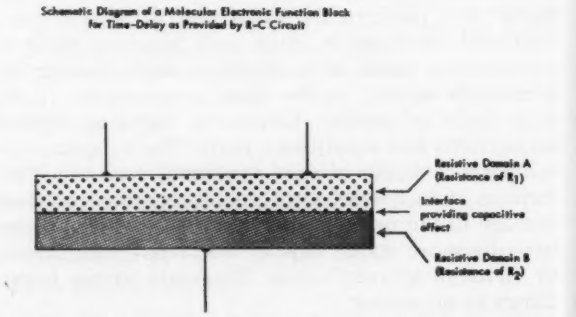
Schematic Diagram of a Molecular Electronic Function Block for Time-Delay as Provided by R-C Circuit
Resistive Domain A (Resistance of R_1)
Interface providing capacitive effect
Resistive Domain B (Resistance of R_2)

Figure 3. Schematic Diagram of a Molectronic Time Delay Circuit.

should find application in military systems in the near future. With their small size and light weight, they seem to provide an almost perfect answer to the astronaut's prayer for more data from a smaller volume and a reduced mass. These considerations would certainly overshadow any possible cost disadvantages. As for application in civilian equipment, one can only guess. Everything depends on the cost of mass produced function blocks. One thing is relatively certain, however. It is quite inconceivable that these units, with their effective regions on the order of a few molecules, will not be used extensively, at least in military systems.

Author's Note:

I would like to thank the Westinghouse Electric Corporation for their help in preparing this article.

26,100 BHP ON BENCH TEST OF THE FIRST 9 CYLINDER FIAT ENGINE 900.S TYPE

Early in December 1960 shop trials were carried out in the Workshops of Fiat Stabilimento Grandi Motori, Torino of the first Fiat engine 909.S (9 cylinder unit of the 900.S type) to be installed on a 38,000 t.d.w. motor tanker for Soc. Elios of the Lauro Fleet group.

The engine, 900 mm. bore and 1600 mm. stroke, is rated for the normal output of 2100 BHP per cylinder and therefore with units from 6 to 12 cylniders the power range is covered from 12,000 to 26,000 BHP which are required for the propulsion of merchant ships of big size.

The engine is of the 2-stroke type, single acting, supercharged and represents the natural development of the large bore Fiat engines (750 and 680 types) which are widely spread over and well known for their outstanding qualities of sturdiness, suitability to run on low grade fuels, reliability in service and low maintenance cost.

The main results obtained during the shop trials of the 9 cylinder engine 909.S, at normal rating and overload respectively, are:

	Normal rating	overload
—Output BHP	19.043	26.100
—r.p.m.	122.7	131.4
—m.e.p. Kg/sq.cm.	7.62	9.76
—specific fuel consumption gr/HP.h	154	165.5
—mean temperature at cylinder exhaust °C	298	343

It is reported that this engine developed the highest horsepower ever attained by a diesel engine.

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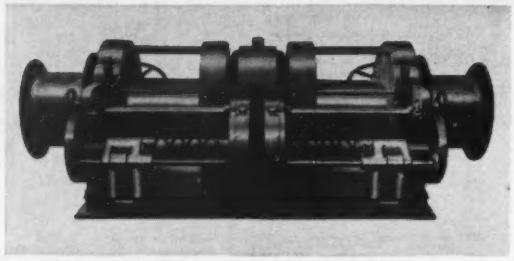
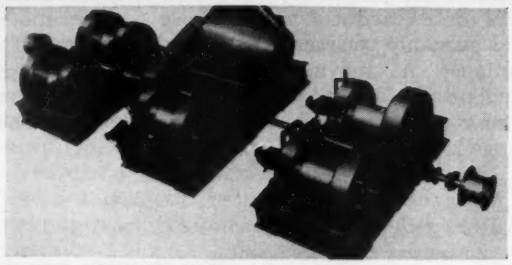
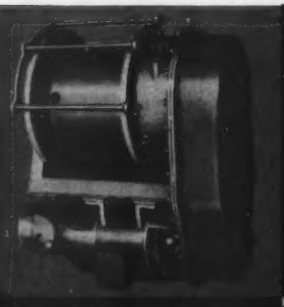


MOORING WINCHES — Single or double drums. Electric, electro-hydraulic or steam power. Automatic tensioning optional. Sizes and horsepower for every application.



DREDGE WINDING GEAR — Drum arrangements to match deck space and application. Sizes and capacity for any dredge requirement.



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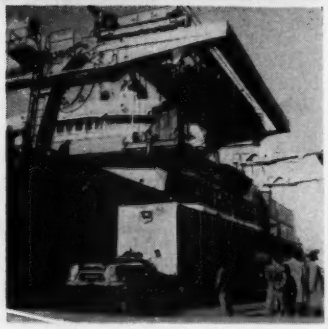
CARGO WINCHES — Electric, electro-hydraulic or steam powered. Capacity and drum arrangements as required by ship rigging plans. Complete line of snaking, stores, hatch cover and ladder winches available.

VERTICAL CAPSTANS — Arranged for above deck mounting of capstan, motor and reduction gearing on common bedplate or with reduction gearing and motor below deck. Capstan head sizes optional. Powered to meet ship specifications. Electric or electro-hydraulic.

WINDLASSES — Vertical or horizontal. Optional electric, electro-hydraulic or steam power. Wildcat sized for anchor chain warping head geared and dimensioned for hawsers to meet ship requirements.

CRANES — Skagit was instrumental in pioneering the container ship crane as illustrated on these Pan-Atlantic ships. Their trouble-free operation attests to the engineering ability, quality and workmanship embodied in Skagit products. Other types of gantry and king post cranes also available for shipboard installation.



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IMPLICATION OF THE LCR MARK II

LIEUTENANT COMMANDER J. C. MEREDITH, USN

THE AUTHOR

graduated from Olivet College, Michigan, in 1934 with a Bachelor of Arts degree. Prior to being commissioned in the Navy he held positions in Wells Fargo Bank, of San Francisco, and with the Treasury Department's Foreign Funds Control section. During the war he served as Financial Censor in the Cable Censorship Office and later as CIC Officer and Fighter Director on board USS STEPHEN POTTER (DD-538). Post-war assignments have included duties as commander of USS HANNA (DE-449), during the Korean conflict and more recently as Staff Officer to Commander Allied Naval Forces Northern Europe. Author of "The Tattooed Man" (Duell, Sloan & Pearce, 1959) and numerous magazine articles. He is presently Executive Officer of Assault Craft Unit One.

". . . A monstrous tripod, higher than many houses, striding over the young pine-trees, and smashing them aside in its career; a walking engine of glittering metal, striding now across the heather; articulate ropes of steel dangling from it, and the clattering tumult of its passage mingling with the riot of the thunder . . ."
H. G. Wells, "The War of the Worlds"

WHERE CALIFORNIA State Highway 75 passes the U.S. Naval Amphibious Base south of Coronado, motorists glancing seaward last year might well have been reminded of this piece of early science fiction. For there on the Silver Strand loomed a steel-legged giant large enough to straddle a house, moving ominously out of the water, bellowing and roaring. It lumbered across the beach and over sand dunes, crushing them like ant-hills, completely impersonal except for a small figure perched aloft . . . perhaps its "brain." People for whom Wells' prophecies have proven uncomfortably accurate must have been reassured to see, at closer range, the words "Army Transportation Corps—LCR Mk II" stencilled on the monster's facade. So there was no panic in Coronado.

In the ensuing months the machine became a familiar sight to all, and was duly reported in the local press as a Landing Craft Retriever being tested by PHIBPAC's Assault Craft Unit ONE, a vehicle especially designed to rescue disabled landing craft from the surf, carry them ashore, and refloat them after they had been made seaworthy.

The design requirement was straightforward

enough, and there is nothing remarkable about its solution: the LCR is simply an oversized gantry crane. However, operational tests of the machine have led to the establishment of certain capabilities not even sought in the first instance, capabilities which may have considerable effect on amphibious warfare.

First, a look at the LCM-8, which is the most modern medium landing craft now in use. It was designed under Bureau of Ships auspices as a large, faster, more seaworthy boat than the old Mark 6. It is a sixty-tonner, capable of slamming ashore at nine knots with a sixty-ton load, double that of the LCM-6. With its 48 per cent larger well deck, the LCM-8 also affords greater versatility in handling combinations of weapons, vehicles, troops, and cargo.

By virtue of its greater size and weight, however, the LCM-8 posed special problems in handling and salvage. Though less inclined to broach than the narrower LCM-6, this expensive boat is by no means immune to disaster, and some kind of retriever had to be devised. A jeheemy-hoist similar to that used with the Mark 6 would not answer the purpose, for this type of carrier is useful only as far out in the surf as a tractor can place it, a clumsy maneuver at best. In dealing with landing craft which draw as much as five feet aft, a shift to self-propelled, deep-wading equipment seemed indicated.

This was the approach adopted by Army's Transportation Research Command ("TRECOM") in 1953 when they assumed the task of developing a suitable retriever. Key figures in the project were Lieutenant Colonel M. E. Manley, TC, USA, and engineers Samuel Hixon and John Sargent in the Marine Terminals Division. A design contract was let, and in due course R. G. LeTourneau (now Westinghouse-LeTourneau) produced a machine capable of handling the LCM-8. This first LCR was a truly monstrous assemblage, over 110 feet long and fifty feet wide, with rubber-tired wheels driven by individual electric motors, and steering by non-tracking articulated means. It worked all right at Fort Storey; but it was stuck there, being virtually untransportable. It is still there.

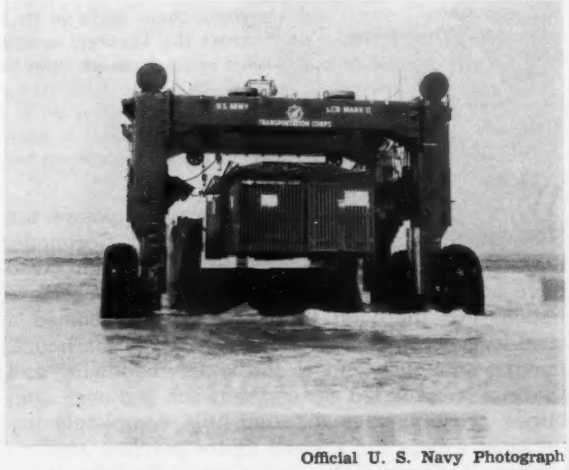
The Army wanted a handier piece of equipment, one which could be demounted and shipped by rail or water. A new design produced by Jerod Industries was executed by Skagit Iron & Steel Works at Sedro Wooley, Washington, and the resultant Mark II had its first acceptance trials in April of 1959. A certain amount of redesign proved necessary, so that the machine was not finally accepted until seven months later. By then it represented a total investment of $270,000.

Meanwhile the Navy adopted the LCM-8 as standard equipment in Assault Craft Unit ONE and

Official U. S. Navy Photograph

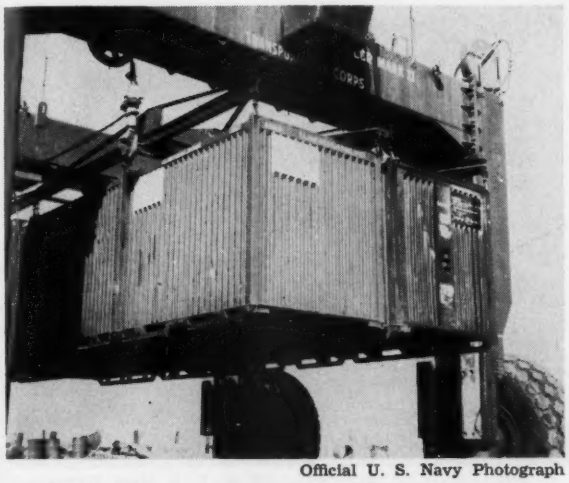
Official U. S. Navy Photograph

LCM-8 in normal carrying position. Total weight of 126 tons can be moved over wet sand, dry soft sand, or rough terrain without difficulty.



Official U. S. Navy Photograph

Carrying CONEX Boxes (45 ton Gross—36 ton net) from LCM-8.



Official U. S. Navy Photograph

Detail of CONEX Boxes and lifting frame after modification. Research engineers complimented the test crews on their home-made frame.

Assault Craft Unit TWO, the organic LCM commands in PHIBPAC and PHIBLANT respectively. This brought about increased interest in a suitable retriever, and the Navy agreed to participate in testing the new LCR. The commanding officer of Assault Craft Unit ONE was designated a member of TRECOM's Project Test Group and was directed to proceed with the operational testing and evaluation of the machine.

Thus it was that in January of 1960 the completed LCR arrived in San Diego on board USS *Carter Hall* (LSD 3) together with numerous spares and a "handy-dandy assembly kit" which alone weighed six tons. It was put together at the test site on the strand, for no one knew if it could cross the highway without tearing up the asphalt like pancake batter.

Although less sinister than the walking tripods described by H. G. Wells, the assembled LCR was an impressive object, rearing its eighty-five tons to a height of amost thirty feet on a twenty- by thirty-two-foot wheelbase. The wheels themselves were nine-foot goliaths driven by two 277-hp gasoline engines built into the longitudinal beams. Control, torque converters, hoist mechanisms, brakes, and wheel suspension were fully hydraulic. There were even two 350-gpm pumps for dewatering swamped landing craft.

The Mark II soon demonstrated its worth as a boat retriever. From any attitude between waterline and nine feet of depth it could snatch an LCM-8 from the surf, bring it ashore, and deposit it on the sand in an average time of nine minutes. Though lacking a steering mechanism as such, the vehicle proved to be quite agile in making skid-turns, with opposing wheels going in opposite directions. Rough terrain didn't faze it, and to see the LCR racing along the beach at ten miles per hour with a 60-ton landing craft clutched to its steel bosom was quite a sight!

Testing continued through the spring months; not always happily. The engines overheated. Pipelines ruptured and spewed expensive hydraulic fluid on the sand. The million-foot-pound brakes faded. Controls broke off in the operator's hand. These and a dozen such ailments as prototypes are afflicted with came to light. But each time the test crew patched up a casualty they became more confident of their machine. It was clear that such a vehicle, after the bugs were eliminated, could accomplish not only ordinary salvage of disabled landing craft but could also make possible a complete system of maintaining, repairing, and storing them without benefit of harbors, cranes, slabs, and other hard installations ashore.

Would the LCR be a useful component of a naval amphibious force? The question hinged on transportability. The Navy traditionally prefers equipment which is ready for immediate action, and if the LCR could be carried intact across the ocean and landed in operational condition its value would be much enhanced.

The possibility had been speculated on by TRECOM. Two suggestions were offered. The first was that the LCR be loaded on two LCM-8 linked abreast, each bearing two wheels. But this proposal floundered because the landing craft, if spaced wide enough to carry the load on their centerlines, could not fit into the well of a Landing Ship (Dock). The second suggestion, namely that the LCR be carried "piggy-back" on a single LCM-8, was much more intriguing: the apparent overload would be largely offset by buoyancy of the immersed wheels, and the center of gravity would be low enough to satisfy the dictates of stability. The overall size of such a

Official U. S. Navy Photograph

LCR and LCM-8 entering well of USS *Thomaston*, 6 July 1960 for transfer to Port Hueneme.

combination, moreover, would be within the well-deck dimensions of some of the LSDs now in service.

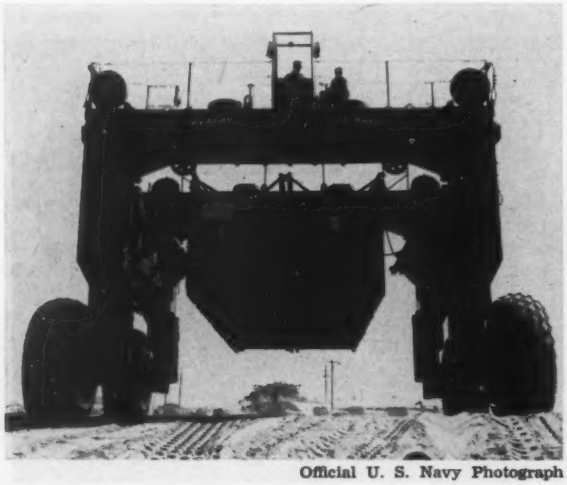
So on March 22, 1960, the LCR hoisted an LCM-8 against special brackets designed to bear evenly on the gunwales, and proceeded out to sea. Not everyone was optimistic: the strange amphibian was ablaze with orange life-jackets, and one man had even provided a buoy marked "Here Lies Mk II." Despite these misgivings the piggy-back combination rolled to deep water, floated away easily, circled once under LCM-8 propulsion, and waded ashore again without difficulty.

A few days later the maneuver was repeated, and this time the LCR was loaded into USS *Thomaston* (LSD 28), then offloaded and beached. Subsequently it was by the same ship transported intact to Port Hueneme 137 miles up the coast, for engineering tests conducted by the Naval Civil Engineering Laboratory.

Meanwile an entirely new aspect of the LCR revealed itself: its potential as a cargo-carrier.

Everyone connected with the World War II landings remembers the tremendous pile-up of cargo at the beach line, the demands for manpower to unload the boats and shift supplies inland, the confusion, the hazardous delays. Elimination of this critical stage from future landings is a prime objective, and post-war efforts in this regard, aimed at projecting cargo *over* the beach instead of *on* the beach, have resulted in developments such as the Army's 60-ton BARC and the Navy's over-the-beach highline transfer system.

The BARC is a special amphibious vehicle, fantastically expensive, and unlikely to replace the ubiquitous LCM. On the other hand, over-the-beach highline transfers may revolutionize offloading of LST and LCU types, but offer less of an advantage in the case of smaller craft due to the high ratio of rigging-time to transfer-time per ton on cargo.



Official U. S. Navy Photograph

View of 45 ton "package" being carried along beach. From this position the load can be transferred directly to low-bed trailers.

Official U. S. Navy Photograph

"Piggyback" combination underway at sea. Submerged portions of LCR afford nine tons of buoyancy.

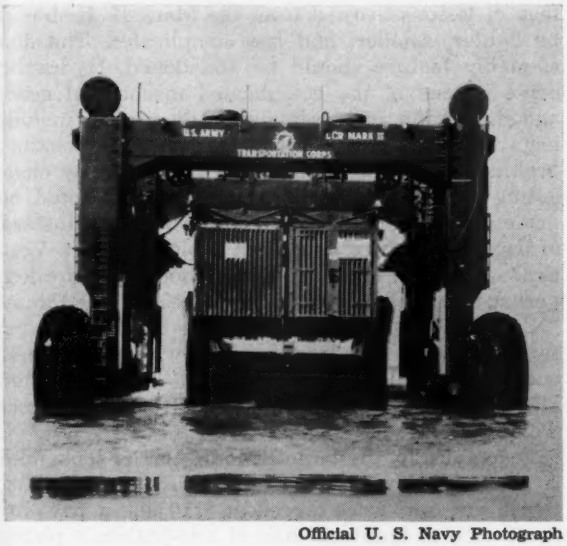
Here the LCR as a prime mover, capable of straddling a loaded LCM-8 and of hoisting seventy tons at a time, promises to solve the problem. Here the only requirement is that the load consist of a unit, or a cluster of identical units, susceptible to simultaneous lift by a sling or frame carried by the LCR: in other words, pallets or standard shipping containers such as CONEX boxes.

In pursuit of this new goal, the Assault Craft Unit ONE test team designed and fabricated a steel frame large enough to hoist eight CONEX boxes at once. The frame, held near its four corners by the LCR's main hoisting hooks, was fitted with thirty-two small hooks for grappling the 344-cu. ft. boxes. The latter were filled with dummy cargo.

When all was ready, the LCR positioned the load in a beached LCM-8, which then retracted. At a given signal the LCM-8 relanded, the LCR straddled it in the surf, hoisted its cargo, and proceeded 400 yards inland to deposit a portion of the load onto a waiting low-bed trailer. The experiment was hindered by lack of suitable quick-acting hooks for use on the boxes. Even so, the LCR succeeded in offloading 39 tons of cargo in the space of twelve minutes!

Additional capabilities were established early this year:

One of these relates to the laying of beach-matting to enable trucks and other vehicles to negotiate in sand, using heavy woven-wire fencing for the purpose rather than conventional accordion-type matting or Marston matting (both of which are



Official U. S. Navy Photograph

CONEX Boxes being removed from LCM-8 with modified frame. 26 May 1960.

much more expensive). Such fencing proves highly "trafficable," and 4000 feet of it, ten feet wide, can be carried and laid by the LCR.

Another capability, related to the machine's tremendous draw-bar pull, permits it to be used to drag the landward end of a pontoon causeway high enough on the beach to be immune from tidal changes.

Still another field of use lies in the landing of POL stocks (e.g. 1200 gallons per CONEX) and potable water.

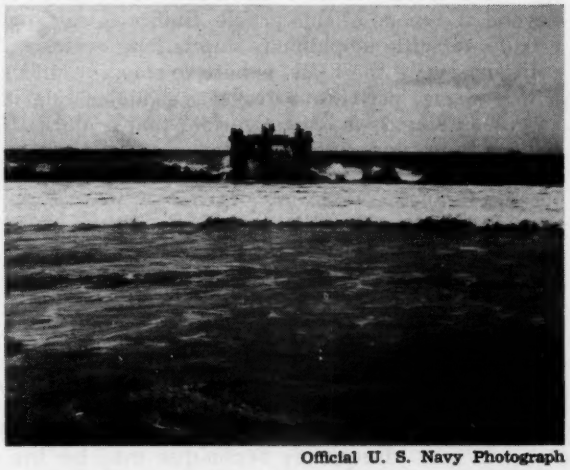
Its usefulness augmented in many ways, the LCR becomes a valuable component in any major landing:

1. For quickly transferring cargo from beached LCM-8 to wheeled vehicles or supply dumps back of the beach.
2. For salvage of disabled and broached landing craft.
3. For miscellaneous prime-mover functions on the beach.

The first of these three functions emerges as the most important of all. At 150 tons per hour, which is entirely feasible, the LCR offers a rate of beachline transfer far superior to any other means now available.

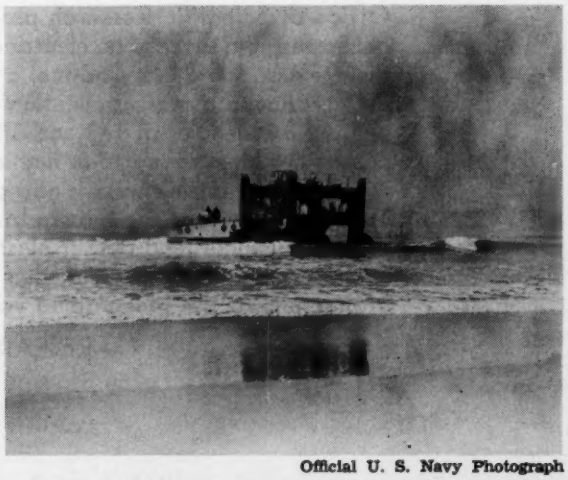
Such acceleration of cargo must have far-reaching effects on the ship-to-shore movement, and gives rise to a new concept of lighterage embodying highly developed collaboration between the LCR and the LCM-8. Commander M. J. Silverman, USN, Commanding Officer of Assault Craft Unit ONE has coined the word "bi-phibian to apply to this combination. In effect it represents a two-piece amphibian. One half, the LCR or similar straddle carrier, would operate in its own proper element, on land. The other half, the LCM-8 (actually several LCM-8) would operate in its proper element, water. Each half would be fully utilized: wheels would not be wasted in the water, and boat hulls would not be wasted waddling inland.

The age of really exotic amphibious equipment is near at hand: hydrofoils, ground-effects machines with air-walls, ground-effects machines with water-walls, etc., . . . but in terms of payload these machines have no early prospect of justifying their immense power requirements, their cost, and the amount of shipping space they would occupy, except for purposes of rushing highest-priority cargo ashore. Until they do, perhaps the "bi-phibian" offers the best means of supporting a large scale landing anywhere in the world.



Official U. S. Navy Photograph

"Piggyback" combination beaching at Coronado, on return from engineering tests at Port Hueneme. Front wheels have just touched down.



Official U. S. Navy Photograph

The LCR prepares to lift an LCM-8 out of the surf. The machine can operate in depths up to nine feet.

In order to exploit to the full the inherent possibilities of the bi-phibian, compatible cargo make-up and compatible ship-design are required.

Unitization and containerization of cargo on a massive scale are inevitable; this has been well pioneered here and abroad, especially among commercial shippers. The CONEX containers adopted by the Army and Air Force are examples of what can be done along this line. The Type 2 CONEX is its own warehouse, is virtually nonpilferable, can be de-humidified, and represents a packaging cost of only 6¢ per pound of capacity. Although technically "returnable," under battle conditions it would more likely be put to such uses as troops shelters, command posts, armories, storehouses, and the like.

Containerization of amphibious cargo implies some rather drastic changes in ship design but the specific proposals under consideration are somewhat beyond the scope of this article. Suffice to say that a truly versatile amphibious ship is long overdue.

In any event, the LCR is here to stay, . . . under a new name, perhaps, to reflect its major role in cargo-handling. It is almost certain that a Mark III will be built, under Navy guidance, to take advantage of lessons learned from the Mark II. It should be lighter, smaller, and less complicated. The disassembly feature should be abandoned. Hydraulic drive instead of the complicated mechanical gear-and-shaft train promises many advantages, including complete compatibility of control, hoisting, braking, suspension, and mobile functions. By elongating the suspension rams the Mark III could be made capable of lowering itself onto a load instead of depending entirely on hoist cables to lift it. Vertical suspension of any size load should be afforded, perhaps through use of movable sheaves fitted in an "X"-frame. As thus envisaged the LCR would resemble a huge, four-legged spider, with engines and operator's cab located in a central "thorax." After all, we can still learn something from the insect world!

Economically, the large straddle carrier looks like a good investment. By saving just two LCM-8, which are now price-tagged at $110,000, it pays for itself. And as landward half of a bi-phiban it places the attainment of significantly higher over-the-beach tonnage rates, which cannot be priced in dollars and cents, within our grasp.

CROSS-SPACE BY UHRF

Acceleration of charged particles to high velocities by a new radio frequency technique may be the method used to propel space vehicles on long interplanetary voyages, according to a report presented to the American Physical Society by RCA scientists. Drs. George A. Swartz, RCA Laboratories, and T. Todd Reboul and Gary D. Gordon of the firm's Astro-Electronics Division, described experiments done in a USAF Office of Scientific Research program. A mixture of ions and electrons generated by successive electrical discharges into a pool of mercury, then released into a cylindrical chamber, was subjected to an electrical field produced by the application of radio frequency power to the chamber. At frequencies of 140 and 330 megacycles, the speed of ions in the plasma has been raised to as high as approximately 40,000 mph, the team revealed, and further work is now planned, at frequencies up to 2,500 mc. They point out that the thrust obtained can be increased by using higher radio frequencies and denser plasma; that the experimental method does not require a magnetic field, thus eliminates need for adding the weight of magnets to the space-going vehicle.

—RESEARCH AND DEVELOPMENT, June 1961

"GAS TURBINE"

A C ELECTRIC DRIVE

ACKNOWLEDGMENT

This article by George E. Pinter, Application Engineer for Jack and Heintz Inc. appeared in the March-April 1961 issue of "Gas Turbine" from which it is reprinted.

LACK OF SUITABLE transmission systems is tending to keep light-weight gas turbines from assuming a more prominent position as prime movers for land and amphibious vehicles. The conventional means for converting the output power of the high speed (20,000 to 40,000 rpm) turbine into usable wheel power (about 100 rpm for large vehicles) can be materially improved.

A-C ELECTRIC TRANSMISSION PROMISING

A "breakthrough" appears very likely in the form of an a-c electric system developed by Jack & Heintz, Inc. A schematic diagram of this transmission is shown in simplified form in Figure 1. The system consists of three basic elements; (a) a-c Generator; (b) Frequency Changer; (c) Induction Motors.

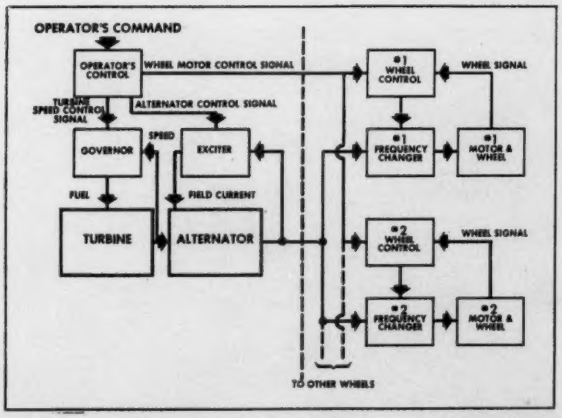


Figure 1. J&H electric transmission system (simplified) with a-c generator, frequency changer and induction motors.

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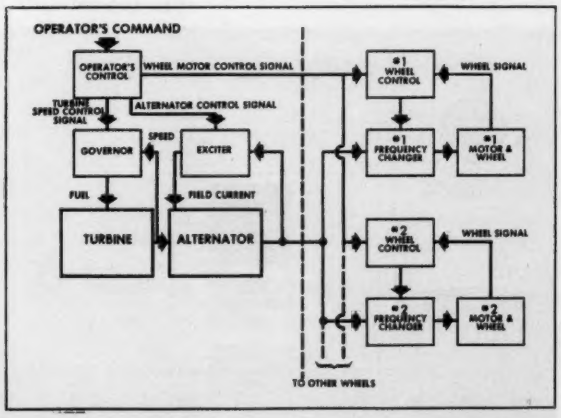

Figure 1. J&H electric transmission system (simplified) with a-c generator, frequency changer and induction motors.

The a-c generator that connects directly to the turbine is capable of handling full output of the turbine. Electric output of this generator is fed to frequency changers from which power is distributed to motorized wheels. The outstanding advantage of this system is that it combines the simplicity, freedom from maintenance, and long life of an a-c system with the almost ideal characteristics of the d-c system.

A brief description of the major elements will aid in understanding the functioning of this transmission.

MOTORIZED WHEEL

Wheel-powering is accomplished with squirrel-cage induction traction motors. This type a-c motor is the most widely used, being extremely hardy and reliable. Also, it is smaller than a d-c motor due to its high rpm capability. Because it has no brushes, slip rings, or commutator, it is possible to encapsulate windings so that the motor can operate even immersed in water.

FREQUENCY CHANGER AND CONTROL

Present form of the frequency changer, which now makes practical an a-c transmission system for vehicles, was developed a number of years ago as a part of the Jack & Heintz Variable-Speed Constant-Frequency power system for aircraft. This VSCF development had as one of its aims, replacement of the normally-used hydro-mechanical device by a highly-reliable static device—an electronic black box. This black box eventually became the static frequency changer.

In essence, this component consists of a number of switches, shown schematically in Figure 2, that are opened and closed at predetermined intervals. In this manner, a controlled frequency wave is fabricated out of small pieces of a number of generator frequency waves (See Figure 3).

This process of opening and closing switches produces a jagged controlled-frequency wave which is

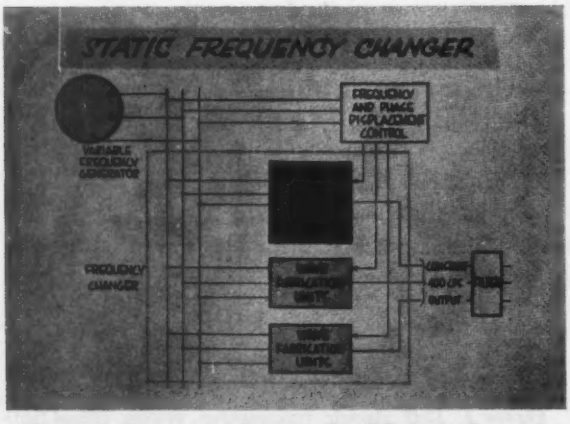

Figure 2. Frequency changer—the "black box"—consists essentially of a number of switches (shown schematically).

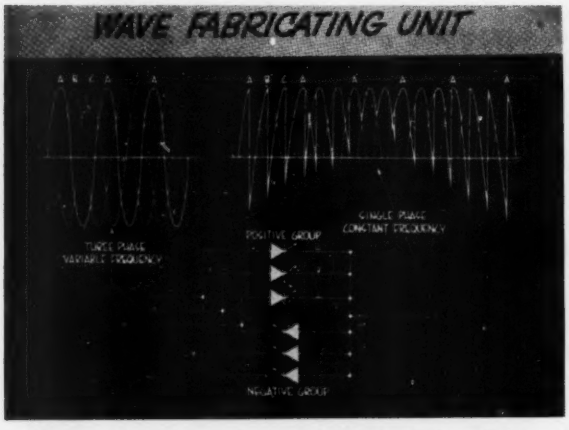

Figure 3. Opening and closing "black box" switches at pre-determined intervals fabricates a controlled frequency wave from small pieces of a number of generator frequency waves.

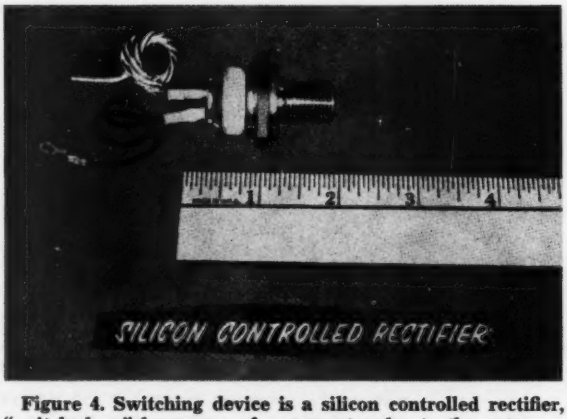

Figure 4. Switching device is a silicon controlled rectifier, "switched on" by means of a current pulse to the gate.

smoothed by means of chokes (or filters). Circuitry is so arranged that frequency changer output is slaved to an oscillator. Hence, output frequency follows precisely the oscillator frequency, regardless of input frequency to the unit.

The switching device in this frequency changer is a silicon-controlled rectifier, Figure 4. This component can be rendered conducting or "switched-on" by means of a small pulse of current applied to the "gate." It ceases conducting or is "switched-off" when the anode-to-cathode current is forced to zero by other components in the circuit. This device is, therefore, the solid state equivalent to a thyratron.

The frequency changer of the type presented here has become a practical reality only since advent of this new semi-conductor device, the silicon-controlled rectifier. Power-handling components of the frequency changer comprise a number of these silicon-controlled rectifiers (usually 18 or multiples thereof). In addition, there are a number (usually 9 or multiple thereof) of firing transformers used to fire the controlled rectifiers at appropriate times.

These firing transformers are, in turn, controlled by a low-power signal (a few watts) from the driver's control station.

A-C GENERATOR

The Jack & Heintz SECSYN (Stationary Exciter Coil SYNchronous) generator, Figure 5, is a unique design of this type machine. This brushless machine has a conventional armature winding located in stator slots and a bobbin-wound d-c exciting winding attached to each of the end-barrels. The rotor carries no windings and it is comprised entirely of steel and other metallic parts, Figure 6. Generator bearings are the only wearing components.

This type construction leads to an extremely hardy and long-life machine. Further, because there are no rotating windings, there is no necessity for rotor cooling. Because no rotating seals are needed, the stationary portion of the generator is adapted to oil cooling.

SYSTEM ADVANTAGES

Advantages of the a-c electric drive system are manifold. Among the most salient benefits are:

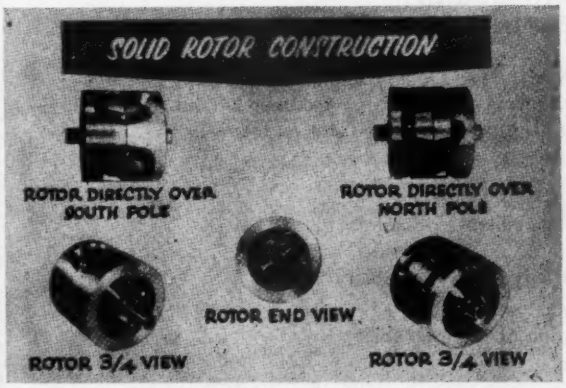

Figure 5. J&H's SECSYN generator has conventional armature winding located in stator slots, bobbin-wound d-c exciter.

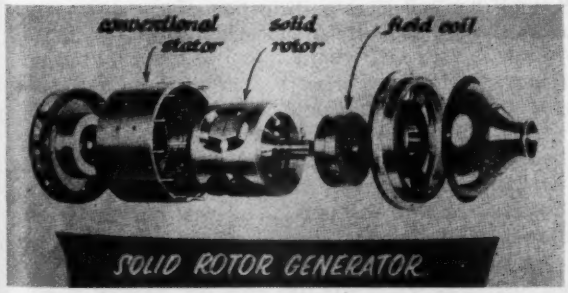

Figure 6. Rotor carries no windings, is composed entirely of metallic parts. Bearings are only wearing components.

Lower Weight: Indications are that significant weight saving can be achieved, especially in the generator.

Control Flexibility: Almost any conceivable scheme for driver control can readily be incorporated into the a-c electric drive system outlined here.

Ideal Speed/Torque Relationship: To achieve best possible vehicle performance, any transmission must deliver essentially all horsepower available from the turbine to the wheels under all conditions of speed. Torque/speed characteristics, Figure 7, apply to the Jack & Heintz electric drive. This system provides an infinitely-variable speed ratio between prime mover and wheels.

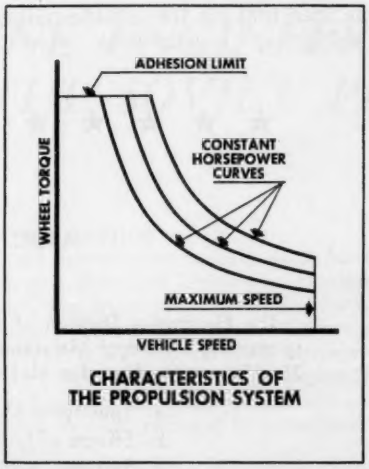

Figure 7. Propulsion system provides infinitely-variable speed ratio between prime mover (power) and driving wheels.

Lower Cost Per Mile: Some facets contributing to this benefit are:

1. As just indicated, turbine speed has no fixed relationship to vehicle speed. Hence, a servo-controlled loop, sensing generator output power, can be utilized in controlling turbine speed. This permits operating the turbine on the *minimum fuel consumption profile* for all conditions of vehicle operation. Operating the turbine at preferential speeds at all times is expected to show significant improvement in vehicle specific fuel consumption.
2. Because components of the a-c electric drive are high-reliability, long-life items, per-mile operational costs will be lower.

Reverse Operation: Reversing is achieved without a gear change mechanism. It is attained by affecting a change in phase sequence to the motors that is accompished by interchanging signal power loads (low power) rather than motor power loads.

Regenerative Braking: This feature is provided in the a-c electric drive system by reducing the frequency changer's output frequency to a value below the synchronous frequency of the motors (produced by wheel rotation). Under this mode, the motors will perform as generators, and the frequency changer, having the capability of bi-directional power flow, will supply power to the generator. The generator, in turn, now functions as a motor, returning energy extracted from the wheels, to the single-shaft turbine compressor unit where it is dissipated.

Regenerative Steering: By using the multiple-unit frequency changer approach, speed of each motor can be separately controlled. Thus, for vehicle turning, the frequency changer setting for the inside-radius wheels (or tracks) can be set at a lower value than that for the outside-radius wheels, thereby achieving regenerative steering. For tracked vehicles, this can be extended to include the pivoting condition.

Powered-Trailer Control: This system is admirably suited for the powering of trailer units. Total caravan power and control is housed in the lead unit.

APPLICATIONS

For the present, it is believed that major a-c power transmission applications will be in those commercial and military vehicles which have off-highway as well as on-highway missions to perform and where flexibility, maneuverability, and reliability are factors of significant consideration. Multi-purpose trucks, personnel carriers, tanks, amphibious vehicles, self-propelled weapons and earth-moving equipment are among the most likely initial applications.

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NOTICE!

ASQC PRODUCT MAINTAINABILITY WORKING SEMINAR

The Electronics Division of the American Society for Quality Control (ASQC) is sponsoring a Product Maintainability Working Seminar to be held on October 24 and 25, 1961 at the Sheraton Hotel in Philadelphia. The program topics are as follows:

a. Specifying Maintainability
b. Effects of Human Factors on Maintainability
c. Systems Level Trade Offs—Reliability vs. Maintainability vs. Availability
d. Choosing Degrees of Automaticity in Maintainability
e. Designing for Maintainability
f. Measurement and Demonstration of Maintainability
g. Predicting Maintenance Time
h. "Throw Away" Maintenance
i. Pricing and Costing for Maintenance
j. Organization for Effective Maintenance

The ultimate aim of the Seminar is to attain firm concepts in the above mentoned areas by promoting the exchange of ideas through the individual participation of attendees representing varied products and interest—both commercial and military. Registration will be limited.

For additional information contact:

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Baltimore 3, Maryland

COMMANDER BERNARD M. KASSELL, USN (RETIRED)

MARINE ENGINEERING NOTES FROM THE SOVIET PRESS

THE AUTHOR

enlisted in the Navy in 1931, and served continuously until his retirement He attended Cornell University and Maryland University, obtaining his B.S. degree in Military Science from the latter institution. He has had a wide range of duty assignments, including tours as Engineer Officer and Repair Officer on various ships and tenders. He was recently stationed in Casablanca, Morocco as Commanding Officer, Military Sea Transport Service and retired after 29 years of active duty. His last duty station was in the Office of the Chief of Naval Operations. Is now engaged in consulting and independent research on Soviet affairs.

THE FOURTH ARM

THE UNITED STATES merchant marine is often referred to as the "fourth arm of defense." As a long time student of the Soviet Union, particularly maritime affairs, this writer wishes we could condition our thinking to bring it into line with Soviet thinking on how best to use the merchant marine not for purposes of defense, but as a peaceful weapon of offense—a militant demonstrator of why our way of life is better than the Soviet way of life.

Our merchant marine is bigger, faster, more modern. Our maritime leaders are well-educated, experienced and shrewd in the face of stern competition from foreign flag operators who challenge our right to legislate for our own protection. Obviously, this is neither the time nor the place to debate the pros and cons of free competition versus subsidies, but, because we are trying to picture for the reader the place that the Soviet merchant marine is beginning to enjoy in the world maritime picture, it is well to remember that there is more to a merchant marine than the bare statistics of ships added to the total, or the amounts of cargo carried, or even the costs of operations.

These are tangible items, readily assimilated in statistical accounts. In this world of today, however, intangibles sometimes count for more than bare statistics, however heartening they may be to the stockholder or to management. Put the price tag, if you can, on the value derived from good will arising because a doctor, during a ship's stay in port, treated local dock workers and their families without charge; or the value of a few seamen cutting cane on a sugar plantation; or the residual impression of documentary films, lectures in the native

tongue of the country and exchange of gifts and letters.

These are intangible items, but they loom large in the small world of the underpaid, underfed and illiterate among the peoples of the emerging nations. And they are not isolated cases. Soviet propaganda is programmed with all the care and attention to detail given to the most precise mathematical calculations. In this case the weapon used is the individual "agitator," or the "agitation brigade," on the individual ship. There are no euphemisms here, only the harsh, but realistic approaches to a tough problem.

There are difficulties, of course. Operating a merchant marine the size of that flying the Hammer and Sickle is no easy task for administrators who are constantly facing problems they have never had experience with. So it is well to remember that in size the Soviet Union still operates a relatively small fleet of ships. It is also well to remember that the fleet is not any more efficient than might be expected of people who take over foreign-built ships with propulsion installations they have no experience with, or people who are placed in command situations without adequate experience at sea because increasing numbers of ships demand increasing numbers of appointments to such billets. There are accidents, damages, problems. Yet the Soviet man has demonstrated a capacity to learn how to handle the tools he is given, whether the tools be mechanical, or propaganda.

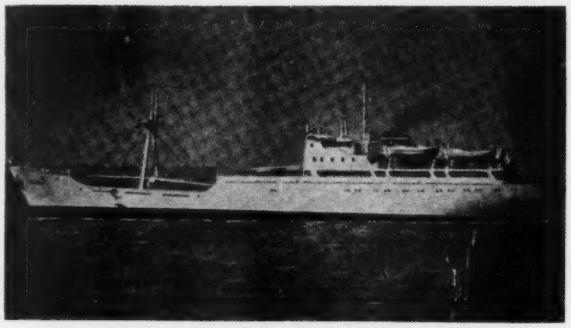
The pay off is, of course, in results. And, from the political and economic points of view, the propaganda and trade offensives throughout the world have been good. Certainly these results are the end product of a series of subsidies, the picking up of the tab, so to speak, today for favors to be received in the future. For example, weigh the cost of providing the Cuban government with a tanker to haul molasses between Cuban ports for a few weeks to help the Cubans distribute the product to storage areas; or similar "lending" of a cargo ship to run between Indonesian ports at a crucial moment in Indonesia's transportation schedule; or the chartering, at a loss, of a ship or two to a needy government to move a perishable or urgent cargo.

The short-term return, at least, is a relatively friendly reception in the ports of call on return trips. In this way fertile ground is available in which to plant more seeds of Communism.

We suggest the use of our own merchant marine as a means of preparing the ground for the planting of some seeds of democracy—US-style.

REFRIGERATOR-FREEZER SHIPS

This series of refrigerator ships, building in the Leningrad area, was started in 1958 and the first of the series, *Tavriya*, was put in service at the end of 1959. Second in the series, *Volzhsk*, came in in 1960, and has been followed by *Sovetskaya Rodina* and *Sovetskaya Rossiya* this year so far. Ships of this



The *Tavriya*-class Refrigerator-Freezer Cargo Ship

type will take the catches from trawlers operating on the fishing grounds, freeze the catches and deliver them to home ports. With an operating time of 60 days and a range of 8,000 miles they will be of considerable assistance to the trawlers. The refrigerators have been built to the Register of the USSR class U1 R4/1S and are single-screw ships with machinery aft. The main characteristics of the class are:

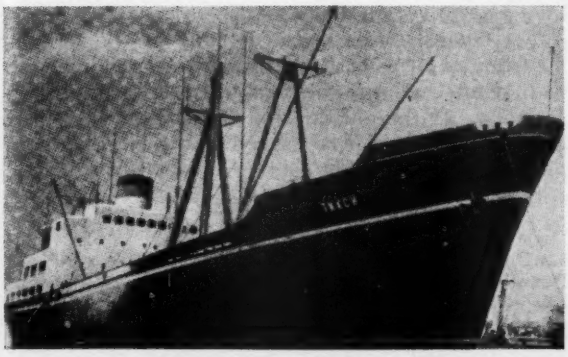
Length over all, feet	326
Length between perpendiculars, feet	295
Beam, feet	46
Height of side to upper deck, amidships, feet	23.6
Displacement, tons:	
Empty	2,900
Full load	5,400
Draft, mean, feet	
at empty displacement	10.5
at full load displacement	18.5
Hold capacity, cubic feet	118,000
Deadweight, tons	2,540
Net registered tonnage	1,190
Main propulsion, ehp	4 x 1,000
Trial speed (at 3800 tons displacement) knots	15
Crew	82

Main propulsion is provided for by the installation of four main diesel-generators Type 5D50, 470 volts direct current, rated at 760 kilowatts each at 740 rpm. The drive motors, two in number, can be connected to the single shaft, switching being such that five different combinations are available for speeds ranging from slow to full power. Voltage to the motors is 920 and maximum revolutions are 180. The main engines, the D50, have proven to be quite reliable in service and have relatively long life, on the order of over 20,000 hours. Fuel consumption is 6.2 ounces per ehp per hour.

Ships of this type were designed for use in the most varied of commercial fishing endeavors and it would appear that this is to be a "basic" type, with changes in design being made to accomodate special purpose use as occasion warrants.

THE CARGO MOTORSHIP *Tiksi*

Tiksi, a general cargo carrier, was built in the Burmeister-Wain Shipyard in Copenhagen, Den-



The Cargo Motorship *Tiksi*, product of the B&W Shipyard in Copenhagen, Denmark.

mark, and delivered to the Soviet Union in mid-1960. Construction was in accordance with Lloyd's Register rules and incorporated ice-use features. It is a single-screw, full-scantling, three-island ship and has been assigned to the Far Eastern Steamship Company for operational control.

Characteristics of the motorship include:

Length overall, feet	518
Beam (less sheathing), feet	64
Height of side to second deck, feet	30
Height of side to upper deck, feet	39
Deadweight, tons:	
Cargo	10,000
Fuel	1,200
Lubes	20
Fresh water	200
Crew and baggage	10
Stores	70
Draft at 11,500 tons deadweight, feet	29
Displacement, tons	17,000
Hold capacity, square feet	570,000
Speed (trials), knots	18
Range, miles	12,000

The five cargo holds are equipped with MacGregor hatch covers, mechanized. 16 5-ton cargo booms and two 60-ton booms, handled with electric winch motors, provide the cargo handling equipment. Each of the 5-ton booms has its own topping winch.

The main engine is a Type 874-VTBF-160 diesel engine built by Burmeister and Wain in 1959, and is rated at 10,000-hp at 115 rpm. Fuel consumption is given as 5.6 ounces per ehp per hour.

RIVER CATAMARAN

Last fall the 40th Anniversary of October Shipyard in Gorkiy, on the Volga River, launched the first catamaran ever built in the Soviet Union. Underway trials on this experimental model were to have been held prior to the freezing of the river. The available photography is indicative of the general outlines of the ship, designed to carry 600-tons of cargo. Decks are said to have been designed to provide for maximum convenience of cargo operations. With engine controls located in the wheelhouse, and with an 11 man crew, the ship is of advanced design and will make some 16 miles per hour.

River Catamaran Built in Gorkiy.

NEW SHIP REPAIR YARD

A new shipyard has been opened in Sovetsk, in Kaliningrad Oblast, the purpose of which is to repair small fishing vessels and to manufacture all types of gear used by the fishing industry. Facilities include a marine railway, hull shop, engine overhaul shop and a joiner shop. The latter facility has been incuded so that repairs to wooden-hulled fishing boats can be carried on.

RIVER GIANT

This proud appellation is given to *Volgo-Don 1*, the 5,000 ton cargo capacity motorship built in the Navashino Shipyard in time to participate in the 1960 river season. First of the class, the ship is of Soviet design and incorporates certain features making it useful not only on the Volga, where water is plentiful, but on lesser endowed rivers, such as the Don, as well. Certain of the characteristics of the class are:

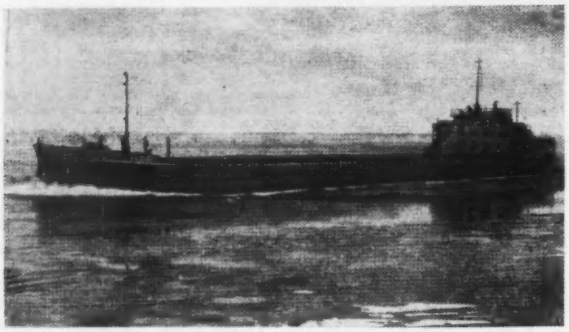
Length overall, feet	458
Beam, greatest, feet	55
Height of side, feet	18
Cargo capacity, tons	5,300 & 4,700
Draft with 5,300 tons, feet	11.5
Draft with 4,700 tons, feet	10.5
Displacement at full load, tons	6,729
Speed, full load and draft of 10.5 feet, knots	10
Crew	18

The ship is designed as a single-decked, twin-screw ship with four open cargo holds. The engine room and living spaces are aft. Built to operate in broken ice, operations call for carrying bulk cargo such as coal, timber, etc., in the Volga-Kama-Don basin, including the reservoirs—and in any type of weather. The hull and superstructure are all-welded, of steel, and the ship is class "O" in the River Register.

Two Mark R8DV-148A diesel engines with turbosupercharging, rated at 1,000-ehp each at 375-rpm, are installed for main propulsion. Directional

shrouds are used as rudders, driven by a sector-type steering engine RER15-3 and an auxiliary rudder is installed in the bow.

Underway trials were conducted on the Gorkiy Reservoir and with the opening of the present season the ship was assigned a routine run. The largest dry-cargo ship on the Volga, *Volgo-Don 1* is considered to be the flagship of the dry-cargo fleet, one which will make it possible to reduce the cost of transporting cargo quite a bit.



The dry-cargo motorship *Volgo-Don 1*, first of a series.

COASTAL PASSENGER SHIP PLANNED

The Ministry for the Maritime Fleet, in its Technical-Economic Bulletin, published the following information on a two-decked motorship with cabins and machinery amidships, a clipper bow and cruiser stern. Main characteristics include:

Length overall, feet	224
Beam, feet	33
Height of side to upper deck, feet	20
Draft, designed, feet	10
Deadweight, tons	305
Net cargo capacity, tons	100
Passenger capacity, berths	110
Crew	34

Aluminum-magnesium alloy will be used for the cabins. The underway qualities of the design were determined from results of model basin tests, using both a towed and self propelled model.

Main propulsion will be provided by two 6DR 30/50 engines rated at 600 horsepower and designed speed is 14.2 knots.

Cargo handing gear will consist of two electric cranes of 1.5 tons capacity each.

LIGHTER TO TANKER CONVERSION

The Yenisey Steamship Line has started a conversion program involving 1,000-ton capacity lighters delivered in the past from Finland. At first it was planned that the 3D12 diesel engine be used for main propulsion, but a later design was approved using two Mark 64RP-25/34 diesels, each rated at 300-hp, providing a speed of about 8-knots. Two of the lighters have already been converted into self-propelled tankers.

THE VOLGOLES-CLASS TIMBER CARRIER

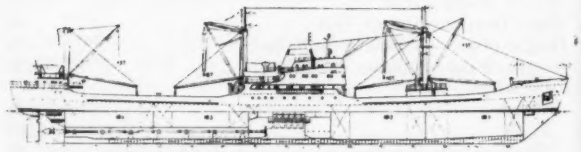
The Gdansk Shipyard, in Poland, has under construction a series of timber carriers for delivery to the Soviet Union, the first of which bears the name *Volgoles*. At this writing four have been delivered and seven are under construction, the most recently announced keel laying having occurred in January.

Some of the characteristics of the class are:

Length overall, feet	407
Length between perpendiculars, feet	378
Beam, feet	55
Height to main deck, feet	27.5
Draft, designed, feet	23
Draft, normal, feet	22.3
Gross Registered Tons	4,600
Cargo carrying capacity, metric tons	6,000
Main engine, horsepower	4,500
Speed, knots (trials)	15.4

Built to class in the Register of the USSR UL*R4/1S, main propulsion is provided by a Sulzer 5SAD72, 5-cylinder diesel engine which will develop rated horsepower of 4,500 at 125 rpm.

Polish designers refer to the class as the B514 type. Designer is Z. Ostrowski in the Central Ship Design Office in Gdansk.



Profile of the *Volgoles*-class of timber carriers, four of which have already been delivered by Poland to the USSR merchant marine.

WHALER BASE *Sovetskaya Rossiya*

October may see the delivery of the newest of the huge whaler bases building in the Soviet Union in the *imeni* I. I. Nosenko Shipyard, Nikolayev. This is the 44,000 ton displacement *Sovetskaya Rossiya*, 714 feet in length and 98 feet in the beam. If the delivery date is met it will have taken the shipbuilders exactly one year from launch to delivery of the twin-screw, 3-decked giant. Dock trials were scheduled for June, obviously a decisive date, in so far as final delivery is concerned.

Addition of this 24,000 ton capacity carrier will bring to four the number of whaling flotillas which will be able to put to sea under the Hammer and Sickle and work the waters of the Far East and the Antarctic. *Sovetskaya Rossiya* will be based permanently in the Far East, as opposed to *Sovetskaya Ukraina* and *Slava*, for example, which are based in Odessa.

RIVER SHIPBUILDING

Modernization of the river fleet operating on the various rivers in the Soviet Union has been under

discussion for two party congresses, decisions of which have led to replacement of obsolete ships, as well as to modernization programs. Thus, at the beginning of 1960 motorships made up 63 per cent of the fleet, a percentage which is to increase to at least 84 by the end of 1965. Wooden barges are to be replaced by steel units. Wooden barges, in 1960, made up 23 per cent of the barge fleet, but all, at least theoretically, are to be gone by the end of 1965.

Whereas cargo motorships accounted for 32 per cent of the annual cargo turnover in 1959, by the end of the present plan period this share in the traffic is to increase to 62 per cent. This will come about by increasing the power and capacity of ships in the various programs, programs which include dry cargo carriers of up to 5,000 tons, tankers capable of carrying 5,000 tons of petroleum products, and sectional barge-pusher combinations carrying from 7,000 to 9,000 tons. Pushing speeds will be increased from 5 to almost 10 miles per hour; cargo ships will be pushed to 12-13 miles per hour and passenger ships will be scheduled at speeds up to 15 miles per hour.

TRAINING SHIP LAUNCHED

The "Neptun" Shipyard, East Germany, was reported to have launched *Gorizont*, a 6,217 tons displacement training ship for the Soviet Union this past spring. According to the reports, the ship is 345 feet in length and 47 feet in beam, equipped with a 3,250 ehp diesel engine which will propel the ship at some 14 knots.

Based on the listed characteristics, and from the past production record of this shipyard, the new training ship is believed to be a converted *Razliv*-class of 3,359 GRT. It will be used to train future mates and marine engineers, and is outfitted accordingly. Radio specialists will also receive their training on board.

Plans call for the installation of a large student wheelhouse equipped with all the modern navigation equipment. Over this wheehouse will be a student bridge with a magnetic compass and peloruses connected to the ship's gyro compass.

The student engineers will have their own classrooms, fitted with machinery most frequently encountered on board present day ships. Included will be a distribution switchboard to handle two 80 kilowatt, alternating current generators driven by diesel engines.

Living compartments have been installed on the main, upper and boat decks, with both students and crew installed in comfortable cabins.

Certain features of the cargo ship remain—for a dual purpose. In addition to providing the students with experience in cargo operations the money earned for freight charges will help to defray part of the expenses of operating the training cruises.

NEW SERIES OF STEAM PROPELLED DREDGER BARGES

The Azov and Black Sea dredging fleet is adding a new series of dredger barges propelled by diesel engines. The first of the series is *Chernomorskaya-1*, said to be very comfortable and fitted with modern equipment. The series will be numbered consecutively, a procedure which is being contested by personnel in the dredging business. These people feel that the policy of naming dredger barges should continue as before; names selected from, for example, capes along the Black Sea, or similar geographic points. The complaint is based on the fact that crews are individual entities, each with a special personality, and one which should not be designated by a simple number.

SHIPLIFTER ON THE NEMAN

Reported under construction on the Neman River, major water artery in the Lithuanian SSR, is an inclined shiplifter designed in the form of a funicular. The purpose of the lift will be to raise ships passing upstream a distance of 66 feet vertically. Ships will enter a huge, steel chamber at the lower level. Water in the chamber will be sufficient to float the particular ship. The chamber will, in some undisclosed fashion, be hooked to the steel rails and lifted to the upper level where it will be deposited in the lake at that point. This type of installation was selected as being considerably cheaper than a system of locks.

250-TON FLOATING CRANE

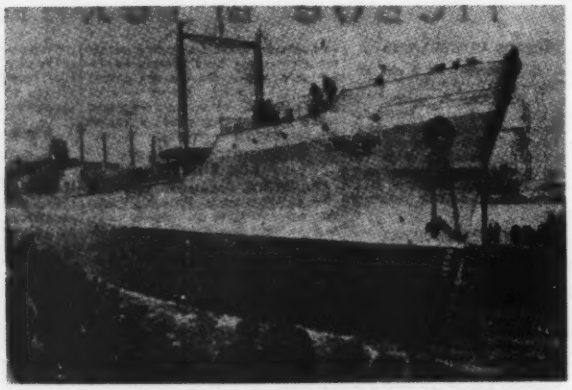
The Leningrad Lift-Transportation Equipment Plant imeni S. M. Kirov has received an order for the construction of a 250-ton capacity floating crane for delivery to the offshore petroleum industry in the Caspian Sea. Maximum lift height is to be 205-feet, the boom reach 215-feet.

TANKERS FOR EAST GERMANY

The Admiralty Shipyard, Leningrad, will produce two tankers for the East German merchant marine this year. The first of these, *Tsayts*, is already afloat and is being completed. The second, *Belen*, was launched on 3 June. This brings to four the number of 10,000 ton capacity tankers built for East Germany in the Admiralty yard.

PORT ICEBREAKERS

Early in May the Admiralty Shipyard announced the launching of the latest in the series of port icebreakers building in the shipyard. Displacing 3,000 tons, the series is a continuation of the *Bogatyir*-class of icebreaker-tug, with certain changes incorporated in the production of the first of the Admiralty-built ships, *Dobryinya Nikitich*, with diesel-electric propulsion.



Tanker *Bohlen*, built in the Admiralty Shipyard, Leningrad, for East Germany at time of launching.

HYDROFOIL PLANS

Present plans call for the construction of 200 *Raketa*-class, and 85 *Meteor*-class hydrofoil passenger ships by the end of 1965, according to a report credited to the Collegium of the Ministry for the River Fleet. This plan is consistent with the recommendation made by the July, 1960, Plenum of the Central Committee of the Communist Party of the USSR.

Interest in this express type of transportation is wirespread in the Soviet Union, undoubtedly partially due to the introduction of the hydrofoil on virtually every major water artery in the country. Analysis of the operations of the craft now in operation, conducted by the Main Administration for Transportation and Operation of the Fleet, as well as by the Main Administration for the Fleet, has confirmed the fact, at least so far as these administrations are concerned, that this type of transportation, for passengers, as well as for cargo, has a great future. Mention of good financial results on runs between Perm and Ust-Kachka along the Kama River and between Irkutsk and Lake Baykal on the Angara River, has undoubtedly contributed to the decision to go into mass production.

However, not all questions connected with operations, and not all the faults so far turned up, have been resolved. One of the major trouble areas is concerned with insufficient study having been made of safety of navigation for the swift craft at night. Yet another deals with inadequacy of between-voyage inspections and maintenance of equipment; central supply of some important parts for the *Raketa* has yet to be organized.

This season, 1961, was to have been one in which careful study of operating areas was to have been made. This year, too, was to have seen aviation principles of use of cargo carrying capacity, as well as engine loading, put into practise.

Perhaps indicative of the manner in which other scientific endeavors have been carried out is the manner in which the hydrofoil program has been driven to fruition. It is only after the fact that we now find that rules connected with technical operations are to be worked out for both the *Raketa* and *Meteor* classes; only after the fact are questions concerned with organizing repair bases to be decided; and still undetermined are questions connected with required types of fuel and lubes, spare and replacement parts and, perhaps most important of all, training of personnel.

REINFORCED CONCRETE FISH FREEZER

A non self-propelled, reinforced concrete fish freezer is under construction in the Gorodets Shipyard near Gorkiy on the Volga River. The ship was ordered by the Astrakhan *Sovnarkhoz* for use on the Caspian Sea. Aiding in the construction are the "Krasnoye Sormovo" Shipyard and the Gorodets Ship Repair and Machine Works, working to complete the ship by the end of this year and have it ready for trials at sea.

According to the shipyard's Chief Engineer, V. Petrov, the floating fish freezer will be considerably different from existing ships used for the same purpose. He anticipates that the use of concrete will give the ship a longer useful life than other types of construction. Although non self-propelled, the freezer base will carry 800 kilowatts of diesel-generator capacity, as well as compressors and the freezing installation. The main deck will be fitted out to receive and process the fish as well as to manufacture ice. The freezer installation is designed to handle 25 tons of fish per day and hold capacity is 300 tons.

CONSTRUCTION OF *Sofiya*-CLASS TANKERS CONTINUES

The Admiralty Shipyard, Leningrad, is continuing its construction of the largest class of tankers ever undertaken in a Soviet shipyard. Designated *Sofiya*-class (the name of the first of the class), two more are planned for placement in service by 1963. It will be recalled that *Sofiya* will displace some 60,000 tons and will be able to carry 45,800 tons of cargo. The hull is 755 feet in length and 102 feet in beam. A 20,000-hp steam turbine propulsion installation will steam the class at 17.5 knots. Mechanization and automation is stressed so that the size of the crew can be reduced to 51. Preparations are well underway, it is claimed, for the construction of the first of the series.

TRAWLER FOR GHANA

Ghana, through the overtures reportedly made by Robert Okran, called a "commercial fisherman" in the Soviet press, will receive an undisclosed number of trawlers built in the Soviet Union. Mr. Okran was a recent visitor to the Soviet Union and during the course of that visit requested plans for a ship suitable for use by his countrymen, with construction to follow.

The Design Bureau of the State Institute For Planning of Fishing Industries, according to Chief Engineer V. F. Golikov, head of the technical phases of the project, got together with the engineers and technicians in the Design Bureau in the "Leninskaya Kuznitsa" Shipyard, in Kiev to draw up the plans for what will be a new type of trawler. The shipyard, it may be noted, is a long-time builder of ocean-going trawlers.

As reported, the ship will be of the medium trawler class (SRT) but will incorporate freezing operations to handle tuna, mackerel and sardines. In addition, equipment will be available for either trawling or purse-seiner operations. Four freezers will be provided to handle the catch, which is to be fresh frozen.

The first of the trawlers to be built will come off the ways in 1962. In the meantime, in order that the fishermen of Ghana can become familiar with the handling of a modern trawler, the Soviet trawler *Ochamchirye,* fishing in the central part of the Atlantic Ocean, was ordered to Ghana to take on board native fishermen for training.

SOVIET DIESEL-ELECTRIC SUCTION DREDGER

Designers from the "Krasnoye Sormovo" Shipyard in Gorkiy, together with personnel from Central Planning-Design Bureau No. 8 have provided plans for the first diesel-electric suction dredger capable of carrying the materials dredged up, the first such by Soviet designers at least. It is planned for use in dredging sandy or silted approach channels to Soviet ports.

The hold capacity for dredged materials has been computed for 1,300 cubic yards; the maximum dredging depth of the suction head will be 49 feet. Other characteristics are:

Length, feet	263
Beam, feet	46
Height of side, feet	16.4
Draft loaded, feet	11.5-13.8

COMMUNICATIONS

May 7th, in the Soviet Union, is Radio Day. Usually celebration of the event is marked by articles on past "triumphs," as well as by those concerning forecasts of things to come. The 1961 event, for example, was a time for an admission that while it was possible for shipping authorities ashore to talk to ships in the Indian Ocean or on the equator, not all ships were so equipped, that while some of the passenger ships assigned to the Far Eastern and Black Sea steamship companies could talk by radio telephone to any city in the Soviet Union, it remained for the future to have radio telephone communications with all the ships in the transport fleet.

The communications industry has been given the go-ahead for the development of a shipboard set and, by 1962 it is expected that a UHF set will be available for installation on some of the ships which will enable them to talk, while in coastal waters, with the ports. In the latter, tie-lines will permit the ships to talk directly to any subscriber on the inland telephone network. A set of this type is planned for all cargo ships.

Industry is to provide, this year, a new relay installation which will provide for reliable, high quality reception from widely dispersed stations and subsequent transmission to the ships concerned. Semi-conductors were used in the manufacture of the equipment.

Considerable attention is being devoted to improvement of emergency equipment. A new model transmitter for shipboard use is soon to be produced by Soviet industry and in production is a "very economical and reliable" semi-conductor emergency radio receiver.

Teletypewriters, using magnetic tape, are planned for use to speed up normal telegraphic transmissions and receptions and wire photos of weather maps will soon be standard on all ships—once equipment becomes available and is installed. First ships to get this equipment will be the large passenger ships and cargo ships.

Two years hence the merchant marine is to begin to get a new version of the "Don" radar, one which will show on the scope both own-ship and other-ship motion. Radar ranges, untended and powered by batteries charged automatically by wind-driven generators, are to be installed in narrows and on shoals over the next two years as well.

While not communications equipment, in the strict sense of the word, gyro-pilots, or "automatic pilots" as they are called by Soviet writers, come within the purview of the Administration for Communications and Electrical-Radio Navigation of the Ministry for the Maritime Fleet. Because units have been installed in ships of the *Leninskiy Komsomol, Pekin,* and *Kazbek* classes, and have given satisfactory service, plans now call for the installation of similar units in all cargo ships "in a short time." A five per cent saving in underway time is anticipated from the use of the gyro-pilot.

There is some discussion with regard to the use of computers to provide the calculations for ships' positions. The idea seems to be to feed the information obtained from prior programming, in the form of coordinates, to the gyro-pilot on a continuing basis leaving to the mate on watch the sole job of making sure all the equipments are functioning properly.

Ashore emphasis is on extending the range of the radio centers and stepping up the traffic flow. Construction of new centers, equipped with modern, automatic gear using efficient, directional antennas, are planned.

Planned for the large ports and ship yards is the use of industrial television. Odessa and Leningrad

have already been equipped with such installations. Mentioned as next in line for such equipment are the ports of Poti, Petropavlovsk-Kamchatskiy, and others.

RIVER SHIP IMPORTS

This past spring a group of river passenger ships, including *Vilgelm Pik, Yuriy Dolgorukiy, General I. D. Chernyakhovskiy* and *General N. F. Vatutin*, as well as the powerful tug *No. 6074*, left Leningrad for the Volga and Kama rivers by way of the inland route to Arkhangelsk thence via the Northern Dvina and Sukhona rivers to their destinations. These motorships, all of the three-decked class, were built in East Germany and represent the continuing contribution of yards in that country to the Soviet economy, at the same time releasing building capacity within the Soviet Union for other purposes. The tug will ultimately operate out of Temryuk on the Kuban River.

EAST GERMAN RIVER PASSENGER SHIP

East Germany has been a consistent builder of river passenger ships, many of which have been exported to the Soviet Union. Most recently reported is an order received by the Edgar Andre Shipyard for nine such craft, the characteristics of which are:

Length overall, feet	175
Length between perpendiculars, feet	166
Beam, feet	25
Height of side, feet	8.3
Draft, feet	8.3
Displacement, tons	361
Seating capacity, passengers	658
Endurance, hours	76
Crew	6

The hull is of steel, divided from bow to stern by five transverse bulkheads, three of which are watertight. Framing is, for the most part, longitudinal.

Main propulsion will be provided by two 4-cycle diesel engines Type RaPV136, rated at 225 horsepower at 380 rpm. The plant is adequate for providing for a speed of some 10 knots. Two 35 kva diesel generator units will provide alternating current lighting and power.

AUSTRIAN-BUILT RIVER PASSENGER SHIP

Some time ago the Soviet Union announced the acquisition of two passenger ships for Danube River service built in the Korneuburg AG Shipyard in Austria. Named *Amur* and *Dunay*, the two were assigned to operate between Izmail and Vienna as tourist ships, under control of the Danube Steamship Company of the Ministry for the Maritime Fleet.

Although the hulls are welded steel, some aluminum was used in the superstructure and remote engine controls were installed in the wheelhouse. The characteristics of this type are:

Length overall, feet	282
Length at the waterline, feet	272
Beam, feet	29.5
Height of side, feet (amidships)	9.5
Draft, feet	5

These craft have a passenger capacity of 240 and will make 18 miles per hour. It takes a crew of 25 men to man a ship, with an additional staff of 83 persons to care for the passengers.

Eight watertight bulkheads and four oiltight bulkheads divide the ship into 13 compartments. Two 8-cylinder, 4-cyle Deutz engines rated at 1200-hp at 380-rpm are used for main propulsion. The steering gear is hydraulic.

There is no indication as to the scope of this program.

SHIPBUILDING IN HUNGARY

Hungarian shipyards, which have been supplying the Soviet Union with cargo ships for some years now, have added to their deliveries 1,200-horsepower pusher tugs and river passenger ships equipped with 800-horsepower engines.

SHIPBUILDING IN CZECHOSLOVAKIA

The government operated shipyard near Komarno, Czechoslovakia, known as the Gabor Steiner Shipyard, was built in 1948 and consisted of eight shops with a total area of approximately 538,000 square feet. New equipment made it possible to build ships under cover and launch them only when maximum degree of completion had been attained. The use of covered shops provided for a maximum use of personnel and equipment throughout the year. The yard, employing some 4,000 workers, has been specializing in the construction of several types of ships of interest to the Soviet Union, designs for which have been worked on in the yard's own Design Bureau, but models, built on the designes, have been tested in the model basin in Vienna.

An assembly-line process of building 50-ton ship sections has been in use and half of all hull welding is done using automatic and semi-automatic processes. X-ray examination of welded seams, on a 10 per cent sampling basis, is said to be standard practice.

Plans are underway to rebuild the slip to take ships of up to 2,500 tons displacement, but at present the largest ship built is a 1,480 tons displacement passenger ship equipped with 300 cabins and capable of carrying 419 passengers The chief characteristics of the class are:

Length overall, feet	316
Beam, feet	38
Beam overall, feet	49
Height of side, feet	14
Draft, feet	8

Main propulsion to the three propellers is provided by three 6-cylinder Skoda diesels, type 6L275, rated at 400-horsepower at 550-rpm. Reduction gearing results in maximum propeller turns of 285 per minute. Developed speed is 13.5 knots. In the bow is a sub-rudder installation which uses the Voith-Schneider propeller and which, as is the case for the main engines, can be remotely controlled from the bridge.

Passenger ships of this type have been welcome additions to the passenger fleet operating on the Volga River, as well as on other rivers in the Soviet Union.

In addition to the passenger ship described above, the Gabor Steiner Shipyard also is building two types of twin-screw, 800-horsepower tugs. The first of the two types is classed as a lake-river tug, and has the following dimensions:

Length overall, feet	158
Beam, feet	29.5
Height of side, feet	10.5
Draft, feet	7.4

Displacement is 518 tons. Two, 6-cylinder Skoda diesels, type 6L275, are used for propulsion and are equipped with reduction gears to drive the propellers which are in tunnels. A tug of this type is said to be capable of towing a caravan of 8 to 10 barges against the current on the Danube River at a speed of 5 knots.

The second type, known as the *Dunaj* (Danube) class, is designed for rescue work and fire-fighting, and uses the same Skoda engine, the 6L275, for main propulsion. Two such engines give the tug a speed of 12-knots running downstream without a tow, or 4 knots upstream with a tow. Principle dimensions of this type include:

Length overall, feet	140
Beam, feet	23
Draft, feet	4.3

In progress are plans for building 1,100-hp tugs.

Other production from Czech yards includes floating pumping stations, the first group, five in all, have already been delivered to the Soviet Union. A model of the unit was made by the producing yard and was presented to the Soviet Ambassador to Czechoslovakia as a gift for Mr. N. S. Khrushchev.

SHIPBUILDING IN RUMANIA

The Galats Shipyard, largest in the country, is presently building seagoing motorships with a cargo capacity of 4,500 tons, as well as various types of river ships. The yard, since the end of the war, has launched 500 ships, according to recent reports. Production of the above type of cargo ship has replaced construction of small river tugs and tankers.

The 4,500-tonner was wholly designed and built by Rumanian engineers and the fifth of them was launched in early 1961. This is the class currently under construction for the Soviet Union. As a result of the agreement reached between the Soviet organization "Sudoimport" and "Industrial-export," the Rumanian foreign trade enterprise, Rumania is to deliver six ships of this class in 1962-63, to be followed by the delivery of 30 lake-river motorships of 2,000 tons capacity each in the period 1963-65.

The Turnu-Severin Shipyard, long a builder of fishing boats, is now said to produce 50 seagoing trawlers annually.

A third yard, in Oltenitsa, is currently building 2,000-tonners.

For the future the planners call for construction of new types of ships, including some as large as 10,000 tons. Much of the production goes to Rumania's own merchant marine, which, it is said, will be able to increase its tonnage carried in 1965 to a figure which will be 7.5 times as large as the 1959 tonnage.

CONSTRUCTION OF FLOATING DRYDOCKS CONTINUES

The Baltic Shipbuilding Yard in Klaypeda is building a second floating drydock which, when completed, will be towed around to the Far East for use of the fishing fleets in the area. The first tow to be completed took 95 days from the Baltic to Vladivostok. The dock, delivered late in May, was designated *MR-5*.

SHIPBUILDING IN POLAND

Shipyards in Gdansk, Stettin and Gdynia were, this past spring, building a total of 60 ships displacing 330,000 tons, all of which are to be placed in service this year. Involved are 24 different types, ranging from 19,000 ton tankers down to small fishing trawlers. Future plans call for timber carriers, bulk cargo carriers, gas-turbine ships, refrigerators and trawlers which will use the stern method of fishing.

SHIPBUILDING IN EAST GERMANY

Shipyards in East Germany are said to have built 2,600 ships of over one million tons in the past 15 years. By the end of 1965 these same shipyards will have built, in particular, 13 fishing vessels, 9 refrigerators, 13 liners of various types, 8 cargo ships of 12,000 tons displacement and 41 ships displacing 4,200 tons. According to press reports these ships are mainly for export to satellite nations. East Germany will, for her own needs, build 20 cargo ships of 11,200 tons, 16 ships of lesser tonnage and some 80 trawlers.

EXPERIMENTAL USE OF GRAPHITED METAL

Engineers of the Leningrad Higher Marine Engineering School and the Central Scientific-Research

Institute of the Ministry for the Maritime Fleet selected the Polish-built cargo carrier *Adam Mitskevich* (3816 GRT) as the site of the experimental use of graphited metal for pistons and associated parts in auxiliary steam machinery. Object of the experiment was to provide parts which would not need lubrication, thus not only reducing the quantity of lube oil required, but also reducing the need for filters and very nearly eliminating carry over of lube oil into the boiler feed water system. It was expected that cylinder wear would be reduced and machinery life extended.

One steam-propelled generator was selected for the trial, with most of the difficulties encountered resulting from poor mechanical qualities of the first rings made for the trial. Improvements appear to have been made in all parts concerned, with the result that, by last spring, the ship had been outfitted with new parts for two steam-propelled generators, two forced-draft fan engines and one steam engine driving a centrifugal pump. Certain of the rings had, at the time of the report, already operated for 4,500 hours and were still in good condition. However, the packing is wearing at an excessive rate, despite lack of wear of cylinder walls and other surfaces, as well as lack of scoring and seizing of rods and stems.

HUNGARIAN-BUILT TOWBOATS

With the opening of the river navigation season transfers of ships which have wintered in various ports take place with increasing frequency. One such was the transfer of the Hungarian-built towboats *Dunayskiy-6*, *Dunayskiy-7*, *Dunayskiy-8*, *Bakhchisaray* and *Gelendzhik* from the Black Sea to Astrakhan on the Volga River via the usual route, including the Volga-Don Canal. It is suggested that the tugs, all diesel propelled and large in size, will be placed in service with the Volga Tanker Steamship Line.

OPTIMUM TRAWLER SPEEDS

The size, and speed, of satellite fishing trawlers has, for many years been a matter of concern to western nations. Particularly important, from a military point of view, has been the question of speed since, all other factors being equal, trawlers are readily convertible into mine layers and mine sweepers.

Trawlers in use in the Soviet fishing fleet today have, on trials, clocked in excess of 13 knots, but it is doubtful if future ships of this type will go much beyond this figure. From the standpoint of the commercial users, where distance to and from fishing grounds, weight of catch, weather and hydrography are all factors, the optimum trawler speeds range from 11 to 12.5 knots. The operators point out that good fishing grounds are distant from home ports and fuel consumption is heavy at the higher speeds. Unless speeds are reduced hull space which might be used for storing the catch goes to fuel storage. The alternative is the use of transport ships plying between the fishing grounds and the base ports, an expensive proposition on its own.

The Polish designers, producers of the *Leskov*-class trawler, have recently confirmed their belief in the under 13 knot design.

SOVIET COMMERCIAL FISHING SHIPS

The following table provides a ready reference to various of the classes of commercial fishing vessels in use in the various fishing fleets of the Soviet Union. Note the joint use of type-number and name designations.

Type	Designation	Name	Built	Dimensions, feet LOA	B	H
Seiner	BChS	—	E. Germany	86	19.7	9.85
Seiner	SO	—	E. Germany	95	20.3	9.85
Seiner	RS-300	—	E. Germany	96.5	20.3	9.85
Seiner	RS-300	—	USSR	110	21.7	11.5
Trawler	SRT (with quarterdeck)	—	E. Germany	129	24	11.2
Trawler	SRT (with poop)	—	E. Germany	126	23.6	11.5
Trawler	SRTR	*Bologoye*	USSR	143	25	12.4
Trawler	SRTR	*Okean*	E. Germany	167	29	14.1
Trawler	RT-22	—	Sweden	200	29.5	15.9
Trawler	RT-161	—	Sweden	206	30.5	15.9
Trawler	RT-10	—	Finland	200	29.5	15.9
Trawler	RT-123	—	Poland	195	29.6	16.3
Trawler	RTM	*Tropik*	E. Germany	262	43.3	23
Trawler	BMRT	*Pushkin*	W. Germany	277	44	24
Trawler	—	*Mayakovskiy*	USSR	278	46	24.3
Fish Freezer	—	*Druzhba*	E. Germany	188	29.5	13.4
Refrigerator	PR-600	—	E. Germany	270	43	22
Trapping Schooner	—	—	USSR	119	26.2	13.8
Trapping Schooner	—	—	Finland	132	30.5	14.7
Whale Catcher	—	*Mirnyiy*	USSR	209	31	17.9

MERCHANT MARINE ADDITIONS

Name	Where Built	Type	GRT
Abakanles, m/v	Poland	Timber Carrier	4600
Alitus, m/v	USSR	Tanker[1]	
Alupka, m/v	Bulgaria	Passenger	
Alushta, m/v	Bulgaria	Passenger	
Argus		Salvage Tug	
Balaklava, m/v	Poland	Tanker/Base Ship	12600
Druzhba, m/v	Poland	Fish Factory	2670
Fizik Lebedev, s/s	USSR	Cargo	12016
Fryanovo, m/v	Finland	Cargo	1869
Gorizont, m/v	East Germany	Training Ship[3]	
Gurzuf, m/v	Netherlands	Tanker[5]	
Gyurtyan, m/v	Bulgaria	Tanker	3000
Kapitan V. Fedorov, m/v		Tug	
Keyla, m/v	Hungary	Cargo	1194
Khoper, s/s		Cargo	
Kotovskiy, m/v			
Kruzenshtern, sail		Survey[4]	
Lgov, m/v		Cargo[6]	
Medveditsa, m/v		Refrigerator	
Mezenles, t/e	USSR	Cargo	3900
Murmashi, m/v		Cargo	
Pavel Chebotnyagin	USSR	Fish Factory[4]	
Petropavlovsk, m/v	East Germany	Passenger	4871
Piltun, m/v	Hungary	Cargo	1194
Snezhinka		Refrigerator	
Sovetskaya Rossiya, d/e	USSR	Refrigerator[7]	
Sovetskiy Azerbaydzhan, d/e	USSR	Ferry[8]	
Tavriya, d/e	USSR	Refrigerator[7]	
Tropik	East Germany	Trawler	
Vanino, m/v		Cargo	
Viimsi		Ferry	
Vilkovo, m/v		Cargo	
Vladivostok, m/v	East Germany	Passenger	4871
Vyiru, m/v	USSR	Tanker	8229
Yukhan Syutistye, m/v	East Germany	Fish Factory[9]	
Yuriy Gagarin, s/s	USSR	Cargo[10]	12016
Zhemchung, m/v	Poland	Fish Factory	2670

1. An 800-ton capacity tanker, first of a series of five under construction.
2. Displacement 6217 tons.
3. Deadweight 24,815 tons.
4. Displacement 6000 tons.
5. Cargo deadweight 7300 tons.
6. Launching reported.
7. See description of type in text.
8. For Caspian Sea service.
9. Displacement 3800 tons.
10. Under construction. The name of the first Soviet "cosmonaut" has been given to many other ships, all on the various rivers. This is the only merchant ship so named thus far.

CORRECTIONS TO PREVIOUS LISTINGS

The following corrections appear to be in order at this time. Details have appeared in the three previous issues of the JOURNAL and these corrections are the result of later, and what appears to be more accurate information.

Name

Arsentyev, d/e vice *Arsenyev*.
Boshnyakovo, s/s vice *Vozhnyakovo*.
Mamin Sibiryak, m/v. Built in Poland vice USSR. 2670 GRT vice 3170.
Mga, m/v. Built in East Germany vice Hungary. 3359 GRT vice 1194.
Ussuriysk, m/v. Built in Denmark vice USSR. 9500 GRT. Cargo vice Refrigerator.
Volzhsk, d/e. *Tavriya*-class. GRT unknown.
Yasnomorsk, m/v vice *Yasnogorsk*.
Yokhannes Vares, s/s vice *Zhokhannes Vares*.

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For further information on the 618S-1/MC, AN/ARC-38A, and other airborne communication equipments write: Marketing Dept., Airborne Systems Division, Defense Electronic Products, Radio Corporation of America, Camden 2, New Jersey.

The Most Trusted Name in Electronics

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THE SOLID STATE PARAMETRIC AMPLIFIER

ACKNOWLEDGMENT

This article was written by Mr. R. D. Weglein, Head of the Solid-State Devices Group in the Electron Dynamics Dept., Hughes Research Laboratories and appeared in the First Quarter 1961 issue of "Vectors," published by Hughes Aircraft Company.

MODERN COMMUNICATIONS systems, earthbound yesterday, today probe the near-zone of the globe with orbiting satellites in attempts to achieve total world coverage. Tomorrow, technologies pointed toward deep outer space may span galactic distances.

Success in these fields hinges largely on our capability to generate and focus vast amounts of transmitting power, and on our ingenuity in devising methods of essentially noiseless signal detection somewhere within the microwave frequency range. Much has been achieved toward these goals in recent years.

The invention of two devices—the maser (VECTORS, VOL. I, No. 1, 1959), and subsequently the solid-state parametric amplifier—has advanced the art of low-noise reception very close to the ideal of completely noise-free detection of microwaves.

Just what is a solid-state parametric amplifier? Most simply stated, it is a device for amplifying signals in the UHF (ultrahigh-frequency) and microwave regions of the electromagnetic spectrum. Why the term "solid-state?" Because the detecting element or "heart" of the amplifier is a semiconductor crystal, no larger than a grain of rice. By its very nature, the diode itself contributes very little noise; hence the signals to be amplified can be very small. It is for this reason that the parametric amplifier has become so important in communications.

An excellent *mechanical* analogy to a parametric amplifier is the ordinary play swing. A child sitting on a moving swing is a simple pendulum whose frequency of vibration depends exclusively on the length of the rope or chain. The total energy of the pendulum is constant so long as the child remains in a fixed position relative to the swing. In trying to increase the "travel" of the swing the child instinctively elevates the weight of its body as it passes over the lowest point in the arc, then lowers it at the highest point—*twice* during each complete swinging cycle. (See Figure 1.) The changes in position are timed so as to *impart* energy to the system on each downswing, giving rise to the term "pumping." As energy is pumped into the system,

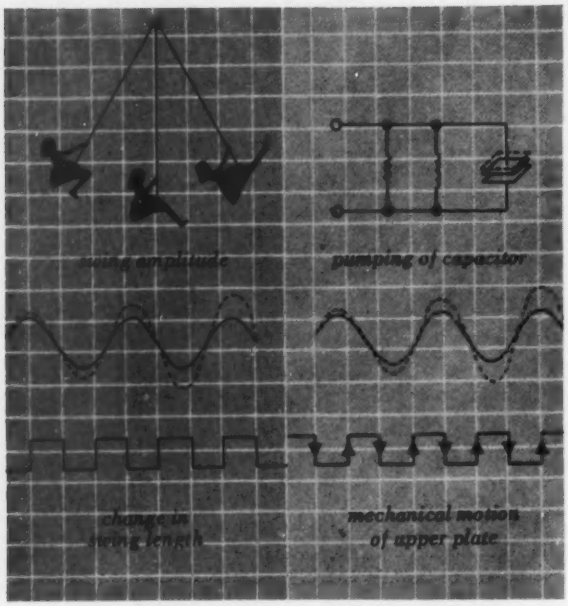

Figure 1. At left, diagram of mechanical analogy to a parametric amplifier—the ordinary play swing. Right, diagram of electrical version at microwave frequencies: analog for the variable-length pendulum illustrated by play swing.

the swinging amplitude continues to increase. By merely reversing the body action and *absorbing* the energy in the swinging motion, the child can come to rest. Thus, the *timing* between the pumping action and the swinging motion controls the direction of energy flow . . . and the swing's amplitude. The pumping action occurs at *twice* the frequency of the pendulum.

This system is a mechanical example of a "degenerate" parametric amplifier. It contains all the fundamental ingredients of which the microwave amplifier is built. Actually, there are really two "swinging" or signal frequencies involved at which energy is supplied by the pump (i.e. the child); but these are identical (or "degenerate") in this special case. Here the variable parameter is the length of the pendulum which the child by its up-and-down motion periodically lengthens and shortens.

In our electrical version at microwave frequencies there exists an almost exact analog for the variable-length pendulum (Figure 1). The changing length of the string finds its counterpart in the variable capacitance, while the mass of the pendulum is represented by the inductance. The swinging amplitude now corresponds to the electrical charge flowing in and out of the capacitor. Figure 1 shows the similarity between the two systems; the same mathematics describes the behavior of both.

Why does the parametric amplifier have the low-noise properties mentioned earlier, which make it so popular as a microwave preamplifier? The answer is that it contains nothing *fundamentally* noisy, unlike an amplifier employing electron tubes, where the random nature of electron emission from the hot cathodes produces large fluctuations in the current, called "shot noise." In our microwave version of the variable capacitor—the back-biased metal-semiconductor gold-bonded diode (Figure 2)—there exists no current flow, and therefore it can be made essentially noise-free. In actual practice a small amount of "thermal" noise is apparent, because the diode is not a perfect variable capacitor and so tends to absorb some of the microwave energy both from the signal source and from the pumping supply.

Many different types of parametric amplifiers using variable-capacitance diodes have been built since the idea was first demonstrated at microwave frequencies. Two particular examples show the principal features. The first is a parametric amplifier developed at Hughes to increase the maximum acquisition range of an airport surveillance radar produced during World War II. Its basic features are shown in Figure 3. Operating in the vicinity of 3,000 megacycles, it amplifies the incoming signal about 100 times and has a nominal bandwidth of about 20 megacycles.

Refering to figure 3, in order to operate as a low-noise amplifier, the variable-capacitance diode must couple together tank circuits at three frequencies: signal, idler and pump. To achieve amplification,

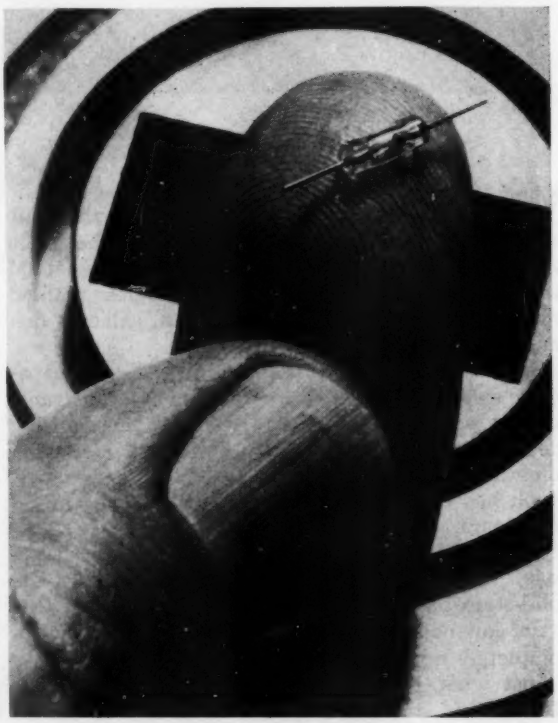

Figure 2. Back-biased semiconductor bold-bonded diode—"heart" of the parametric amplifier.

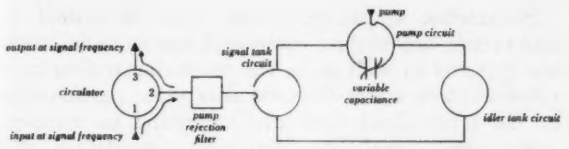

Figure 3. Schematic drawing of parametric amplifier, showing that diode must couple together tank circuits at three frequencies: signal, idler and pump.

the sum of signal and idler frequencies must be equal to the pump frequency. To achieve low noise, the idler frequency should be much larger than the signal frequency. Unfortunately, there is an upper limit for tthe idler frequency beyond which the amplifier becomes more noisy. This restriction comes about because of the imperfect nature of the variable capacitor. The pump serves to modulate the variable capacitance at the pump-frequency rate. The incoming signal is mixed in the varying capacitance with the pump frequency. In a manner analagous to a pendulum, power is extracted from the pumping supply and added to the signal, resulting in amplification. The enhanced signal leaves the amplifier through the "circulator," which ensures proper separation of incoming and outgoing signals, as the directions of the arrows in the diagram indicate.

The modernized radar surveillance system incorporating the Hughes parametric amplifier recognizes signals 6 to 7 times weaker than the earlier version. Because of the vastly improved sensitivity, the range is increased by almost 70 per cent. Significance of this advance can be better understood by scrutiny of the "before-and-after" photographs taken during actual operation of the system.

Pictures shown here represent the received radar pulses reflected from some object (presumably an approaching aircraft) in the presence of normal noise primarily generated within the system itself. Figure 4a ("before") shows the relative magnitude of the received radar pulse, almost indistinguishable from the noise prior to the inclusion of the new parametric amplifier. In Figure 4b ("after") the reduced noise generated within the new device—but brought up to the same level for comparison—amplifies the received signal pulse to show up prominently against the general level of background noise. The compactly designed Hughes parametric amplifier is today in full production.

The second example of advanced parametric amplifier technology is a development by H. R. Buchanan of the California Institute of Technology, Jet Propulsion Laboratory. It served as both a low-noise component and a stable, reliable "building block" in the first practicable satellite communication system, now well-known as the NASA "Project Echo." The 100-foot-diameter Echo I, initial man-made satellite in the program, was fired from Cape Canaveral August 12, 1960, and began orbiting at a 1,000-mile altitude, circling the earth once every two hours.

West Coast terminal of the Echo system is located at the Jet Propulsion Laboratory's Deep Space Tracking and Communications Station at Goldstone, California. Because of the reliability requirements of the Echo system, there were many strict demands placed on the preamplifier. For example, to achieve low-noise performance the amplifer had to be as close as possible to the receiving dish antenna, capable of withstanding severe temperature changes of the California desert, and able to deliver stable, unattended performance in all-weather

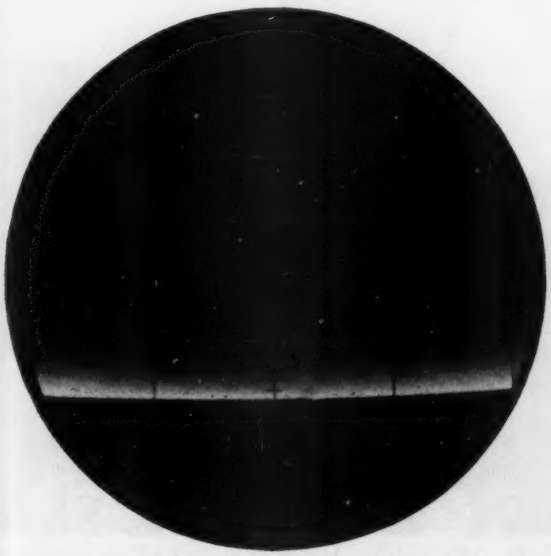

Figure 4a.

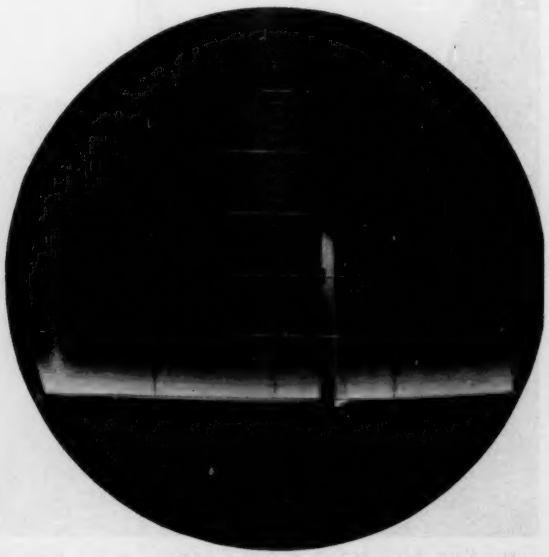

Figure 4b.

environment. Further, it had to be rugged and simple to operate. Electrically, it was required to be sensitive to very weak signals and to be phase-stable so it could be used in a phase-locked loop system.

Essentially, all these requirements were met by the final version of the amplifier installed in the antenna housing of the Goldstone tracking station. The system amplifies signal at 960 megacycles with a bandwidth of approximately 5 megacycles. In order to maintain its gain stability, the pump source is feedback-stabilized to deliver several hundred milliwatts at approximately 9,600 megacycles. The overall noise produced by the amplifier is close to five times the noise entering the input due to random noise from the atmosphere. Incoming noise is usually defined in terms of a temperature in degrees Kelvin (°K)—in this instance measured at 50°K. The thermal noise from a resistor is directly proportional to the ambient temperature. In these terms, the effective "noise temperature" of the Goldstone parametric amplifier is about 235°K.

While a nimbus of uncertainty is characteristic of the early stages of research and development of any new device, certain trends already allow qualitative assessment of the future. Since the key component of the parametric amplifier is the semiconductor diode, advancement of the art is predicated on the state of semiconductor development.

Parametric amplifiers of the type described in this article are finding more and more use in existing systems as well as in the more demanding new ultrasensitive space-directed receivers, particularly in the latter field they are beginning to compete with maser amplifiers. Scientific speculation will undoubtedly continue as to which contender eventually will win the low-noise race.

Clearly, to date, and probably because they operate at almost absolute zero temperature, masers hold the record for low-noise reception. Yet it can be shown mathematically that the maser and the parametric amplifier are quite similar in certain respects; in fact, if both are at the same temperature they have similar lower noise limitations. Purely practical considerations will force a decision as to the ultimate applicability of either device.

The late Professor William W. Hansen of Stanford University, originator of the linear electron accelerator, postulated a theorem which might well be applicable to the foregoing: "If there are two ways of achieving a goal, and both are equally likely to work in principle, then *that* one will be superior on which the maximum amount of development effort is expended."

* A portion of this article presents the results of one phase of research conducted at the Jet Propulsion Laboratory, California Institute of Technology, under the sponsorship of the National Aeronautics & Space Administration, Contract No. NASw-6.

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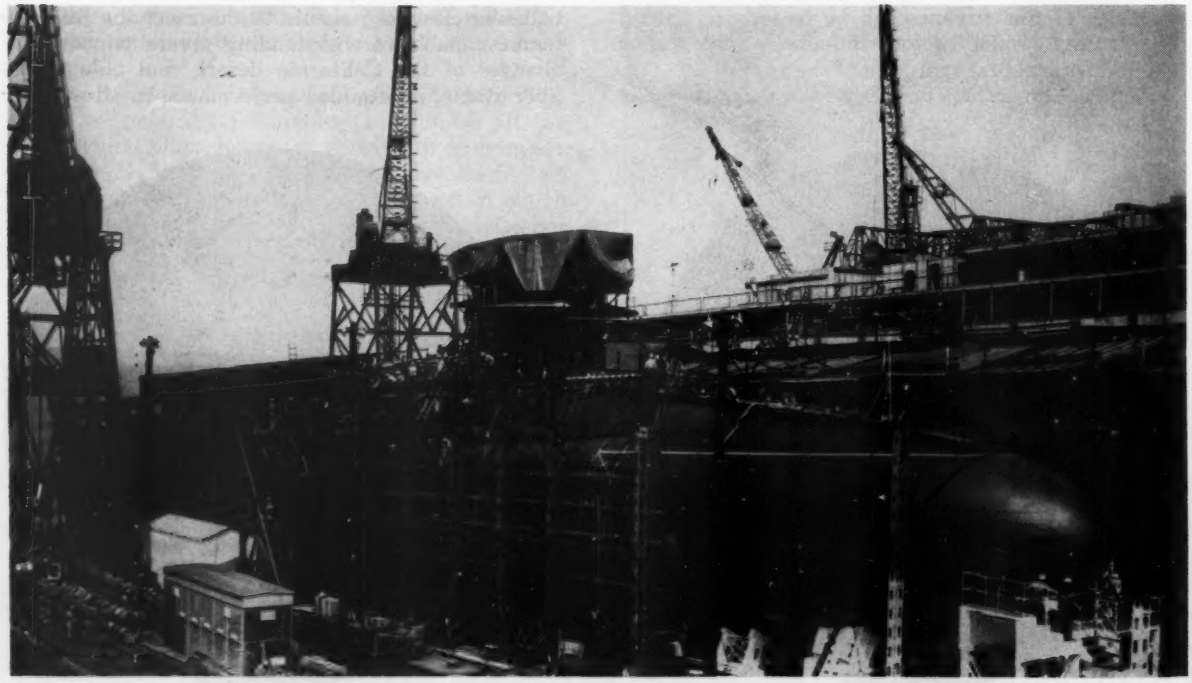

The USS *John Marshall*, latest addition to our country's growing fleet of Polaris submarines, is readied for her launching at the Newport News (Va.) Shipbuilding Plant.

TETSUO NISHIYAMA

AIR-DRAWING AND VENTILATING FLOW CHARACTERISTICS OF A SHALLOWLY SUBMERGED HYDROFOIL SECTION

THE AUTHOR

studied Naval Architecture at the Department of Naval Architecture, Faculty of Engineering, Tokyo University, Japan, from 1944-1947. After several postgraduate courses, he has become an Assistant Professor in the Department of Mechanical Engineering, Faculty of Engineering, Tohoku University in Sendai. He received the degrees of Doctor of Engineering from Tokyo University. His work is connected with general hydrodynamical problems in Naval Architecture, in particular with the submerged hydrofoil and screw propellers.

INTRODUCTION

As a peculiar phenomenon of the existence of the free water surface, this paper will discuss air drawing and ventilating flow together with wave-making.

Up to date, air drawing and ventilating flow has been treated in relation to the problems of the marine screw [1, 2, 6], rudder [7] and submerged hydrofoil [8]. It may be divided into two main types:

The one resulting from the fact that the low pressure region in the tip free vortex of the horizontal wing of finite span plays the part of the pipe conducting the air.

The other resulting from the fact that the dead water space of the vertical hydrofoil of finite span, which is under atmospheric pressure, contacts the atmosphere directly.

However, the author has experienced that an entirely different type of air drawing and ventilating flow from the above two can be seen in the shallowly submerged hydrofoil section; the air drawing and ventilating flow can still occur in the purely two dimensional flow, in which the dead water space does not contact the atmosphere and also the tip vortex does not arise.

Such type of air drawing and ventilating flow has never been examined so far as the author is aware, notwithstanding that it is a most fundamental one.

Under these circumstances, observations have been made in detail and the essential nature has been considered and discussed in conformity with the observed facts; the main problems treated are the condition, the scale effect and the criterion for occurrence of the air drawing and ventilating flow.

TEST EQUIPMENT AND PROCEDURE

The rough arrangement of an open flume used is shown in Figure 1; a flume of 100mm width and 600mm depth was chosen to obtain a relatively high velocity and still insure that the boundary layers would not extend to the pressure taps. And also the two contracted parts before entering the measuring part were chosen to secure uniform velocity distribution and low turbulence at the measured section. The main features of this equipment are as follows:

Since the flume depth is fairly large compared with the foil chord length, the flow velocity can be taken over a wide range of variation below the critical velocity.

In order to avoid the generation of a standing wave in the contracted part which influences the fluid motion in the measured section, this part was closed over to the after edge of the second contracted part.

The hydrofoil sections are an NACA 4412 and ogival section of thickness ratio 12 per cent. It is made of brass and finished as smoothly as possible. The length of chord and span is 100mm, which is equal to the flume width to secure two dimensional flow. The thirty pressure taps—fourteen in the pressure side and sixteen in the suction side—are arranged around the midsection of the foil span, the position of which is shown in Figure 2.

After the preliminary experiments for confirming that the uniformity of the velocity distribution and horizontality of the flowing water surface is practically satisfactory, as shown in Figure 3, the procedure adopted is as follows:

By fixing the wooden weir plate 8 to the outlet of the channel and by raising the foil out of water by lifting the movable plate 5 ,

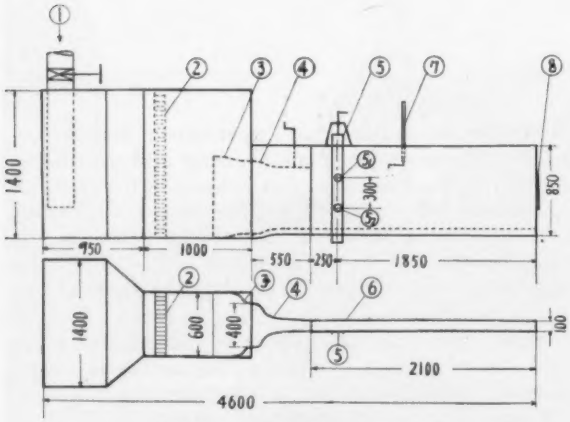

Figure 1. Test equipment.

1. pump used (22 HP); 2. straighter; 3. 1st constricted part; 4. 2nd constricted part; 5. vertically movable brass plate. 5_1. hole used for fixing the hydrofoil section tested; 5_2. hole used for fixing the Pitot tube; 6. window for observation; 7. point gauge (movable); 8. wooden weir plate.

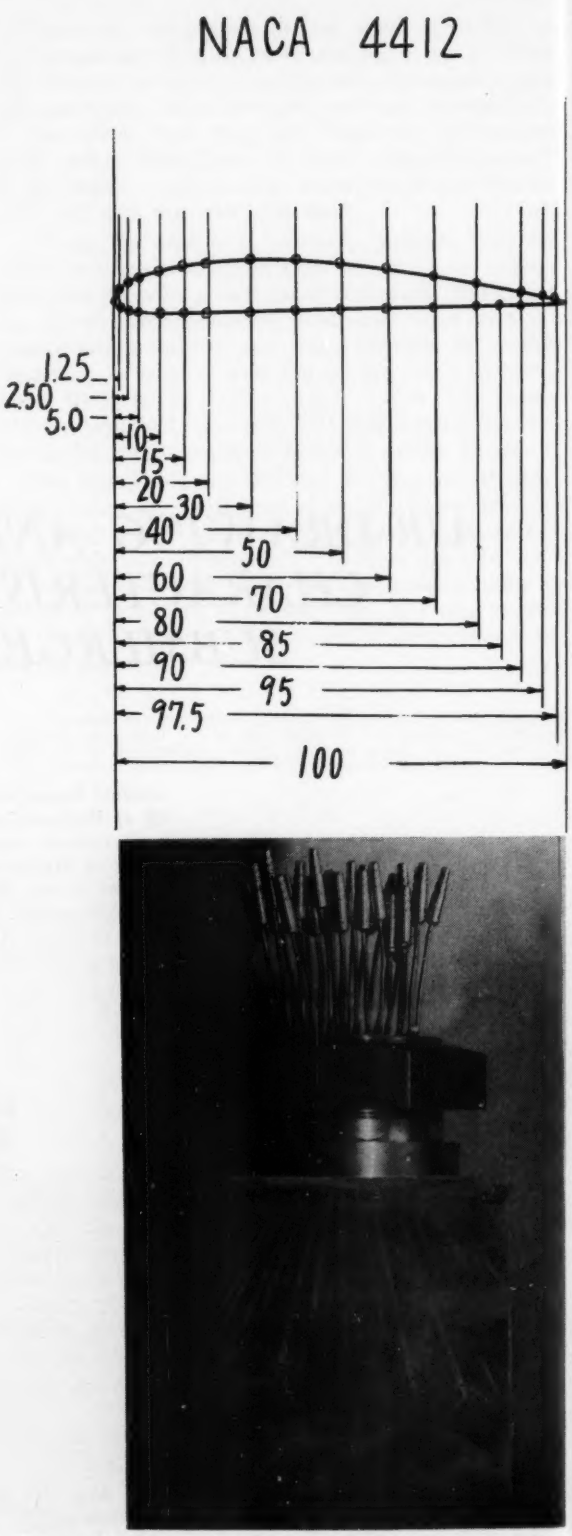

Figure 2. Position of pressure taps and photograph of hydrofoil section.

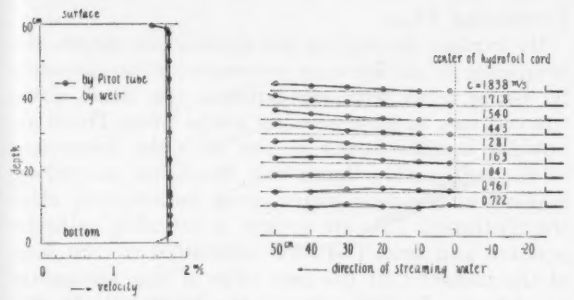

Figure 3. Velocity distribution and flowing water surface at measured section.

to which the foil is attached, we find that by fine adjustment of the pump speed the flowing water surface of the discharge becomes almost horizontal when measured by the scale. The volume of this discharge can be obtained from Rehbock's formula by using the weir of square notch, and the mean value over the measuring section is assumed to be the velocity.

Lowering the movable plate 5 to the required depth of submersion and clearing the air bubble in the rubber tube which connects to the manometer, manometer readings are taken after assuring that it has settled.

Further in order to hold the velocity of the flowing water constant during a test run, special attention is paid to avoid variation of the pump discharge. Figure 4 is an example of the deflections on the manometer board.

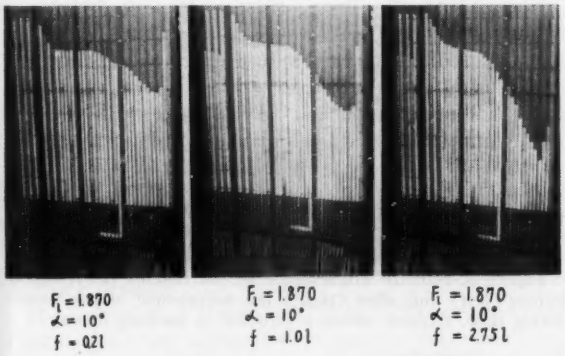

Figure 4. Deflections on manometer board.

Changing the depth of submersion while holding the velocity and angle of attack constant, observations were made and, if necesary, pressure distributions were measured.

OBSERVATION ON THE AIR DRAWING AND VENTILATING FLOW PHENOMENA

Air-drawing

When decreasing the submersion depth, gradually, while keeping the angle of attack and velocity

SYMBOLS

f = vertical distance from still water level to the quarter chord point of the foil
C = velocity of the streaming water
l = chord length
α = angle of attack
g = gravitational acceleration
T = surface tension
η = surface elevation
χ = direction of the streaming water
C_l = lift coefficient
C_d = drag coefficient
C_p = pressure coefficient
F_l = Froude number $(C/\sqrt{gl})$
R_v = Reynolds number (Cl/v)
W = Weber number $\left(C\sqrt{\frac{Pl}{T}}\right)$
v_a = Approaching velocity
v_d = Diverging velocity

of the streaming water constant, the air began to be sucked down into the aftward space of the dead water of the hydrofoil in the form of a bubble. This bubble was carried away by the streaming water before reaching the space immediately after the back of the hydrofoil. Again the next air bubble was sucked down and the same process was repeated. This is just the beginning of the air drawing; a peculiar noise was heard at the instant of the air drawing and the water surface became concave and unstable in the region of sucking down which is discernable with the splash generated.

The critical curve is shown for the beginning of the air drawing in Figures 5 and 6; from these we

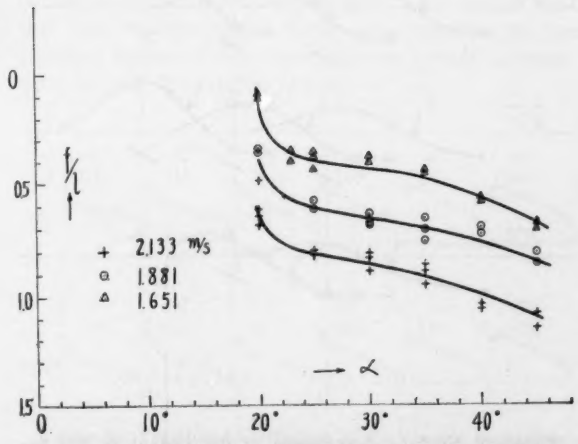

Figure 5. Critical submergence depth for the beginning of the air drawing (Aerofoil section).

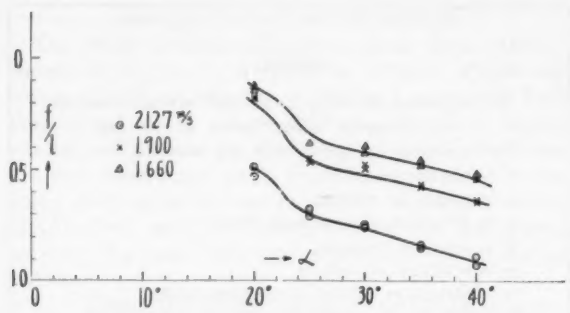

Figure 6. Critical submergence depth for the beginning of the air drawing (Ogival section).

can safely say that the air drawing can be seen only when the angle is larger than the stall angle, and also with the ogival section it is more difficult to suck the air down than with the airfoil section.

The relative position of the surface disturbance to the hydrofoil and the position of sucking down at the instant of the beginning of the air drawing is shown in Figure 7.

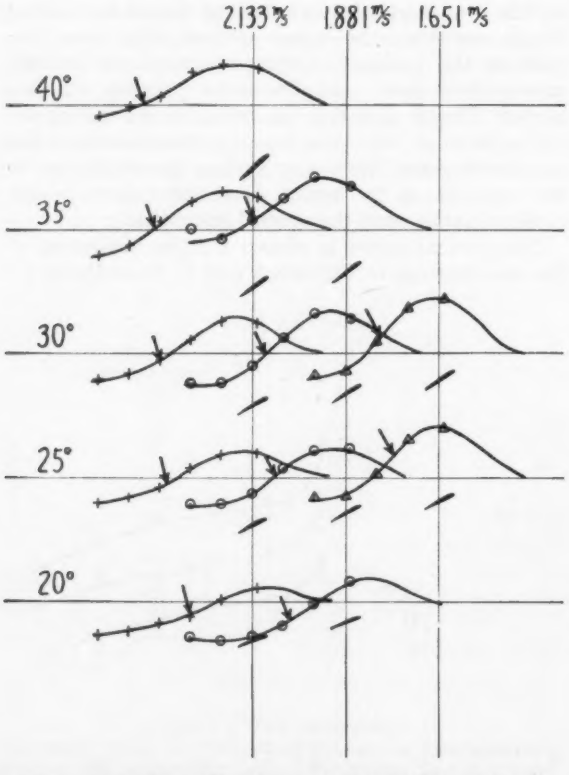

Figure 7. Surface disturbance at the instant of the beginning of the air drawing. (Arrow shows a position of sucking down)

Ventilating Flow

By further decreasing the submersion depth, the frequency of air drawing increases and consequently, being filled with air bubbles, the dead water space comes to have a milky white color. These air bubbles become larger in size with the decrement of the submersion depth and finally an air pocket appears in the dead water space immediately after the hydrofoil. This air pocket is unstable as to its position and size. The flow separation can be seen at the instant that the fore edge of this air pocket touches the leading edge of the hydrofoil. At this time the back of the hydrofoil is covered with air at atmospheric pressure. This is a ventilating flow. The surface disturbance, the position of sucking down and white color region are shown in Figure 8 at the instant immediately before and after the beginning of the flow separation.

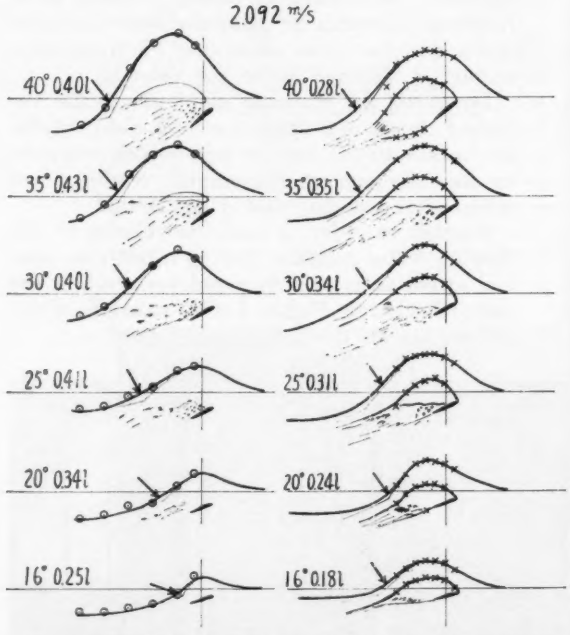

Figure 8. Surface disturbance at the instant immediately before (left) and after (right) the occurrence of the ventilating flow. (Arrow shows a position of sucking down)

Form of cavity

The cavity is filled with air at atmospheric pressure, and the free streamline is stable only in the region immediately after the hydrofoil. As shown in Figure 9, the form of cavity may be regarded as parabolic and can be given from Bernoullis' theorem

$$-\rho g y+\frac{1}{2}\rho A\overline{V}_0^2=\text{constant} \quad \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

where y and $\overline{V}_0$ are: vertical distance from the water surface and velocity at an arbitrary point on the free streamline, respectively.

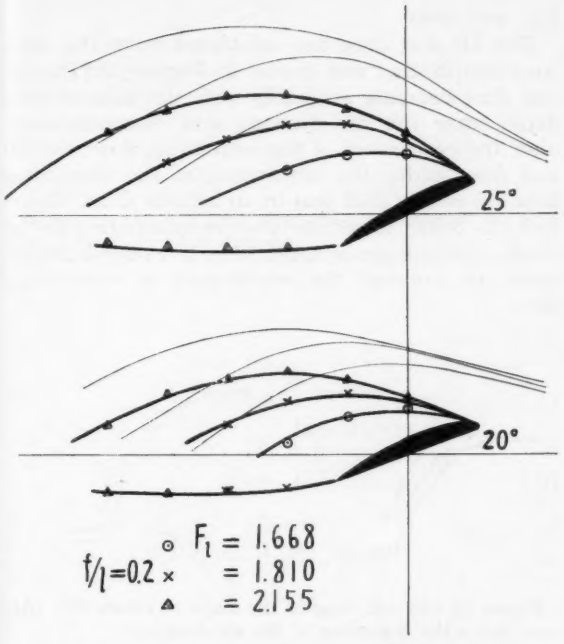

Figure 9. Form of cavity in ventilating flow.

Reforming (1), we obtain

$$-\frac{y}{l}+\frac{1}{2}AF_l^2\left(\frac{\overline{V}_0}{C}\right)^2 = \text{constant} \quad \ldots\ldots (1a)$$

From this we can see that the form of cavity depends on the Froude number, as is clearly shown in Figure 9.

NECESSARY CONDITIONS FOR OCCURRENCE OF VENTILATING FLOW DUE TO AIR DRAWING

Now we will consider the conditions necessary for the occurrence of ventilating flow as determined from the observations. This is of practical importance for foreseeing and preventing the occurrence of ventilating flow.

Surface tension

It is clear that there should be a lower pressure region than the atmospheric one in the streaming water as long as the air is sucked down. Generally the physical relation of the pressure in the two sides —air- and water-sides—of the water surface is given by [9]

$$P_a - P = T\frac{\partial^2\eta}{\partial\chi^2} \quad \ldots\ldots (2)$$

Then, it is necessary to establish the following relation for sucking the air down:

$$P_a - P \geqq 0 \quad \text{i.e.} \frac{\partial^2\eta}{\partial\chi^2} \geqq 0 \quad \ldots\ldots (3)$$

This means that the water surface must be concave upward; in other words, we can say that the equilibrium of the surface tension to the atmospheric pressure breaks and it is possible for the air to be sucked down at the concave region of the water surface.

The surface elevation by the shallowly submerged hydrofoil of normal angle of attack is convex upward above the hydrofoil and concave upward abaft the hydrofoil. Therefore it may be said that the location of sucking air down is just the water surface immediately after the hydrofoil. In fact, this is clearly shown in Figures 7 and 8, where a peculiar splash was observed. Thus the local curvature of the water surface plays an extremely important role and this results from the surface tension only.

Stall

It is necessary to consider another condition for the occurrence of flow separation due to air drawing. Observation shows that the flow separation is closely related to the stall of the hydrofoil. The pressure distribution measured is shown in Figure 10 at the instant immediately before and after the occurrence of ventilating flow. This result shows up in the surface disturbance in Figure 8.

In order that the air sucked down may reach the vicinity of the back of the hydrofoil, the surface pressure on the hydrofoil naturally must be less than the atmospheric pressure and at the same time the dead water space must be moderately deep. This is a reason why stall of the hydrofoil is necessary for the occurrence of flow separation. In fact, as clearly shown in Figure 8, the ventilating flow due to the turbulent flow separation can barely be seen at the extremely small submergence depth while, on the contrary, that due to laminar flow separation can be seen easily at the moderately large submergence depth.

Further, the quantity of the air sucked down in the dead water space must be moderately large for occurrence of the flow separation. This means that the dead water region must be accessible to the concave region of the water surface which is necessary for air drawing to occur.

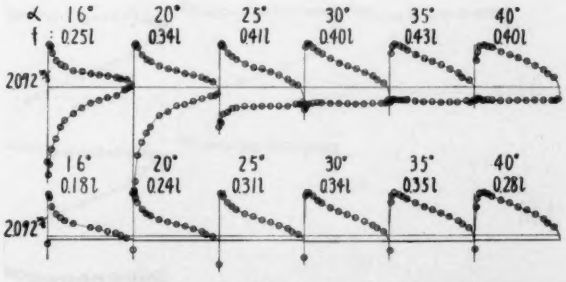

Figure 10. Pressure distribution at the instant immediately before (upper) and after (lower) the occurrence of the ventilating flow.

In short, the necessary and adequate condition for the occurrence of the ventilating flow due to air drawing is that first the hydrofoil is stalled and second the submergence depth becomes so small that its dead water space approaches the concave region of the water surface immediately after the hydrofoil.

SECTION CHARACTERISTICS AFTER THE OCCURRENCE OF THE VENTILATING FLOW DUE TO THE AIR DRAWING

Now we will consider the effect of ventilating flow on the section characteristics.

Pressure distribution

An example of the pressure distribution measured is shown in Figure 11. With the decrement of the submergence depth, the flow rate passing over the hydrofoil decreases gradually and then, correspondingly to this, the absolute value of the pressure decreases. The pressure distribution does not change extremely by the air drawing itself, but abruptly by the ventilating flow, in which the back of the hydrofoil loses instantaneously the greater part of the negative pressure and is under atmospheric pressure. In such condition, only the face of the hydrofoil is in contact with the streaming water.

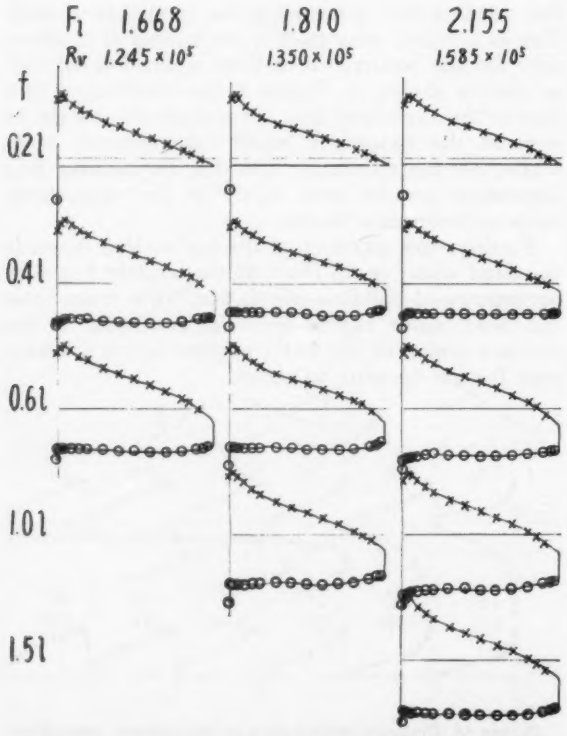

Figure 11. Pressure distribution at various submergence depths in the angle of attack 30°. (Aerofoil section)

Lift and drag

The lift and drag are calculated from the pressure distribution and shown in Figure 12. The lift and drag decrease gradually with the submergence depth after the air drawing and discontinuously after the occurrence of the ventilating flow; the lift and drag under the occurrence of the ventilating flow are nearly half that in an infinite fluid. Therefore, in order to secure the characteristics in the shallow submergence depth, it is of extreme importance to prevent the occurrence of ventilating flow.

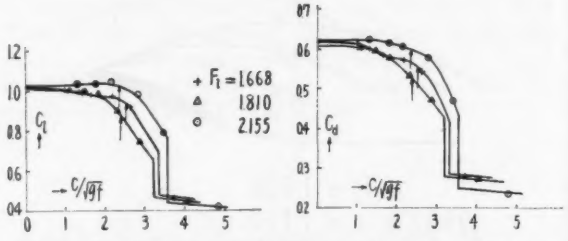

Figure 12. Lift and drag at the angle of attack 30°. (Arrow shows the beginning of the air-drawing)

Observation shows that the form of cavity approaches the final one with the Froude number. Corresponding to this, the lift and drag after the occurrence of the ventilating flow are expected to have a limiting value independently of the Froude number. In fact, as shown in Figure 13, we can see that the Froude number should be more than 2.5, in order to escape the scale effect on the characteristics of the NACA 4412 section after the occurrence of ventilating flow. It is fortunate that this critical Froude number can be easily passed over in the usual experimental tank test.

Comparison with the discontinuous flow theory

Ventilating flow may be regarded as belonging to the category of discontinuous flow. According to Green's exact solution [10], the normal force of the flat plate under the occurrence of ventilating flow is given by

$$\frac{\frac{\rho C^2}{2} l^2}{N} = b(b-\sqrt{b^2-1}) + \frac{b}{\pi \sin\alpha}\left\{2a+(ab-1)\log\frac{b-1}{b+1}\right\} \quad (4)$$

where

$$a=\cos\alpha,\ e^{-i\beta}=\frac{1-ab+\sqrt{(1-a^2)(1-b^2)}}{a-b} \quad (5)$$

Since the direction of the ventilating flow (β) cannot be explained by this theory, it is necessary in calculation to determine it by measurement. An example of comparison is shown in Figure 14; generally speaking, the observed value is about 80 per cent of the calculated one. The main causes for this discrepancy may be stated as follows:

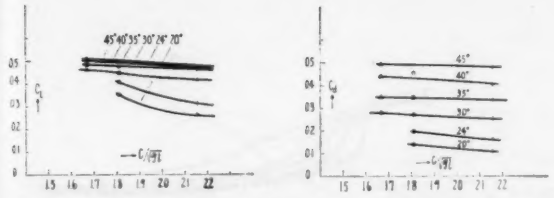

Figure 13. Lift and drag after the occurrence of the ventilating flow (Aerofoil section)

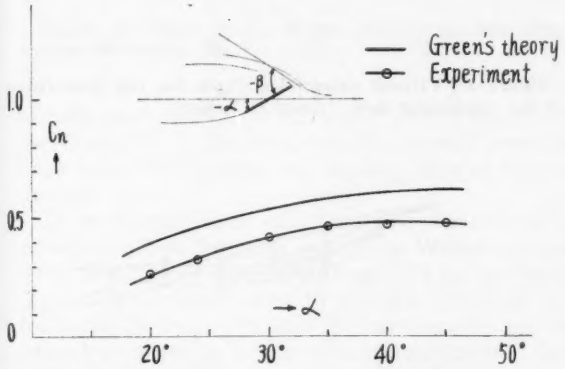

Figure 14. Comparison of the measured value with the discontinuous flow theory.

The point of flow separation does not exist in the leading edge itself, as in the flat plate, but immediately after the leading edge in the hydrofoil of thickness and camber.

The measurement of β is difficult with accuracy and so the mean value is taken necessarily.

The velocity of the streaming water is insufficient for comparison with the discontinuous flow theory, in which the inertia force of the main stream is fairly large.

SCALE EFFECT

Since observation shows that scale effect plays an important role in the occurrence of ventilating flow, we will discuss the details.

Effect of the approaching velocity

When keeping the velocity of the streaming water and the angle of attack constant, the critical submersion depth of the occurrence depends upon the approaching velocity to the free water surface. In fact, as shown in Figure 15, we can see that the occurrence delays with the increase of the approaching velocity. This can be explained by the fact that the air has not sufficient time to be sucked down in the dead water space of the hydrofoil. Hence, in order to make the ventilating flow occur precisely, it is necessary without fail to hold steady equilibrium. This can be done by sufficiently drawing the air, correspondingly to the pressure distribution of the hydrofoil at every submersion depth. From this the approaching velocity should be made as small as possible as can be seen.

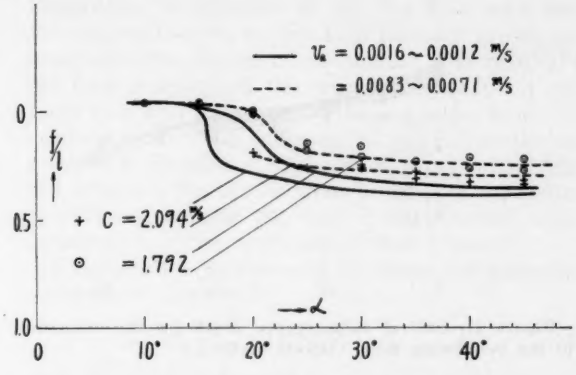

Figure 15. Effect of the approaching velocity on the occurrence of the ventilating flow. (Aerofoil section)

Effect of the diverging velocity

After making ventilating flow to occur, on the contrary, the critical submersion depth for extinction depends upon the diverging velocity from the free water surface. As can be clearly seen in Figures 16 and 17 for the aerofoil and ogival sections respectively, extinction delays with the increase of the diverging velocity. This results from the fact that the air sucked down in the cavity has not sufficient time to be carried away by the streaming water.

Comparing these two figures, we can see the distinct differences as follows:

The extinction depends considerably upon the velocity of the streaming water in the

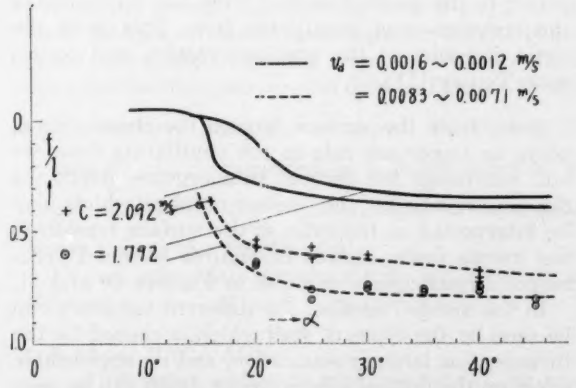

Figure 16. Effect of the diverging velocity on the extinction of the ventilating flow. (Aerofoil section)

FLOW CHARACTERISTICS OF HYDROFOILS

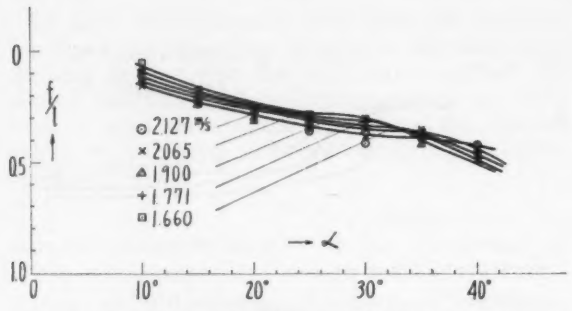

Figure 17. Critical submergence depth for the extinction of the ventilating flow. (Ogival section)

aerofoil section but, only to a small degree in the ogival section.

The extinction is rather hard in the aerofoil section but, easy in the ogival section.

Hence the ogival section may be said to be superior to the aerofoil section from the view point of extinction of the ventilating flow.

Scale effect

Keeping the approaching velocity as small as we possibly can, the critical submersion depth for the occurrence was measured and showed in Figures 18 and 19 for the aerofoil and ogival sections respectively; the critical submergence depth tends to approach the final value, regardless of the sectional form, with the increase of the velocity of the streaming water.

Comparing these two figures, we can see the distinct differences as follows:

In the angle of attack of the region of turbulent separation, no appreciable differences can be seen.

In the angle of attack of the region of laminar separation, the occurrence is rather harder in the ogival section than in the aerofoil section.

Hence the ogival section may be said to be superior to the aerofoil section from the viewpoint of the occurrence of ventilating flow. This is in the same direction as the previous results, and agrees with Tinney [11].

Now, since the surface tension, as above stated, plays an important role in the ventilating flow, we will rearrange the critical submergence depth for this occurrence by the Weber number which may be interpreted as the ratio of the surface tension to the inertia force; curves in Figures 18 and 19 correspond respectively to those in Figures 20 and 21.

In the aerofoil section, the different tendency can be seen by the types of stall which is caused by the turbulent or laminar separation, and no appreciable effect on the critical submergence depth can be seen above the Weber number 80.

In the ogival section, though there is some irregu-

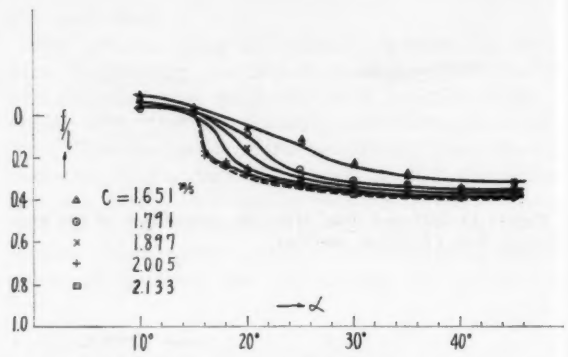

Figure 18. Critical submersion depth for the occurrence of the ventilating flow. (Aerofoil section)

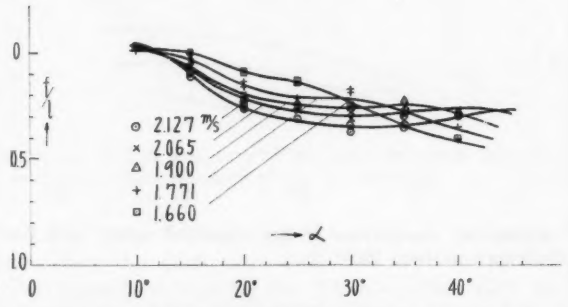

Figure 19. Critical submersion depth for the occurrence of the ventilating flow. (Ogival section)

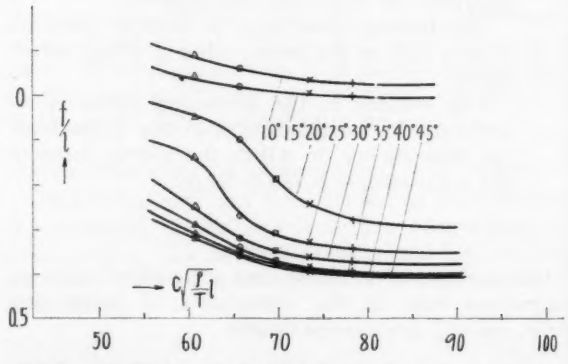

Figure 20. Effect of the Weber number on the critical submersion depth. (Aerofoil section)

lar variation in the low Weber number which results from the breaking to the upstream side of the locally high disturbance just over the sharp leading edge, the same tendency as in the case for the laminar separation of the aerofoil section can be seen and no appreciable effect on the critical submergence depth will be found in the Weber number 100 which is obtained indirectly by extrapolation.

These differences of the critical Weber number

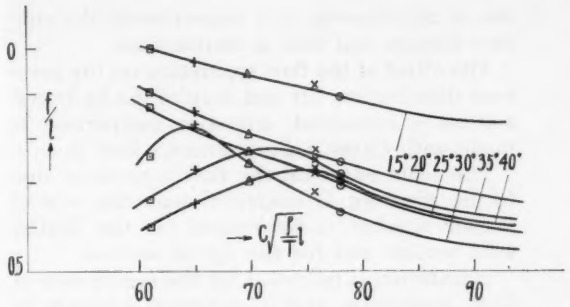

Figure 21. Effect of the Weber number on the critical submersion depth. (Ogival section)

of both sections can be explained by the distinctive features [12] of the pressure distribution over the back immediately after the leading edge of the hydrofoil.

In short, the scale effect for the occurrence of ventilating flow becomes extinct at Weber number 80 in the NACA 4412 section and 100 in the ogival section of thickness ratio 12 per cent. Further it is fortunate that these critical Weber numbers can be passed over easily in the usual experimental tank test.

A DECIDING METHOD FOR THE OCCURRENCE

Now we consider a hydrofoil held in a streaming water and confine our attention to the filament passing over the hydrofoil, i.e., above the dividing streamline, as shown in Figure 22. The force by static pressure and momentum is respectively given by

Pressure $$p_1=\int_0^{f_1}(p_a+\rho gh)\,dh=\left(p_a+\frac{1}{2}\rho gf_1\right)f_1 \quad (6)$$

$$p_2=\int_0^{f_2}p_a\,dh=p_af_2 \quad \ldots\ldots (7)$$

Momentum $$M_1=\rho\overline{V}_1^2f_1 \quad \ldots\ldots (8)$$

$$M_2=\rho\overline{V}_2^2f_2 \quad \ldots\ldots (9)$$

Further, from continuity, we have

$$\overline{V}_1f_1=\overline{V}_2f_2 \quad \ldots\ldots (10)$$

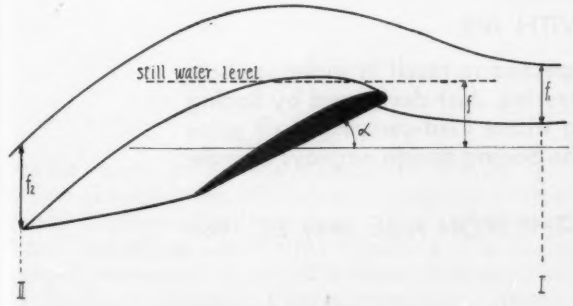

Figure 22. Notations.

Regardless of whether or not the flow separates, the external forces on this field filament are in balance with the change in momentum that results. If the flow is separated, the reaction of the foil is confined to a small part at the leading edge. Since the reaction is the only influence of the foil on the approaching filament considered, it must follow that the criterion for the occurrence of flow separation is a critical value of the ratio of this reaction to the momentum of the approaching flow filament.

This reaction is given by R (from the pressure-momentum theorem):

$$R=(p_1-p_2)+(\underline{M}_1-\underline{M}_2) \quad \ldots\ldots (11)$$

then, this is transformed to

$$R=p_a(f_1-f_2)+\frac{1}{2}\rho gf_1^2+\rho(\overline{V}_1^2f_1-\overline{V}_2^2f_2) \quad \ldots\ldots (11')$$

Photographic observation shows that without the loss of accuracy, the approximation

$$f_1\approx f_2 \text{ is valid} \quad \ldots\ldots (12)$$

then, (12) gives

$$\overline{V}_1\approx\overline{V}_2 \quad \ldots\ldots (13)$$

Therefore (11′) is reduced to

$$R=\frac{1}{2}\rho gf_1^2\approx\frac{1}{2}\rho gf^2 \quad \ldots\ldots (14)$$

On the other hand, this reaction may be given by

$$R=\frac{1}{2}\rho C^2l\times Q\alpha^2 \quad \ldots\ldots (15)$$

where Q depends on the shape of the leading edge. therefore we have, from (14) and (15)

$$\alpha=A\frac{\frac{f}{l}}{F_1} \text{ in radians}$$

where

$$A=\sqrt{\frac{1}{Q}}$$

From this, we can see that the critical angle of attack depends on the Froude number, submergence depth and shape of leading edge. In order to check the considerations above, the data in Figures 18 and 19 are replotted by $\frac{f}{l}/F_1$, as shown in Figure 23; α may be clearly regarded as a linear function of $\frac{f}{l}/F_1$, at least in the region of laminar separation which causes the regular stall.

From a mean line we can decide the constant as follows:

A=3 for NACA 4412
=4 for ogival section of thickness ratio 12 per cent

Therefore we can make use of these curves for de-

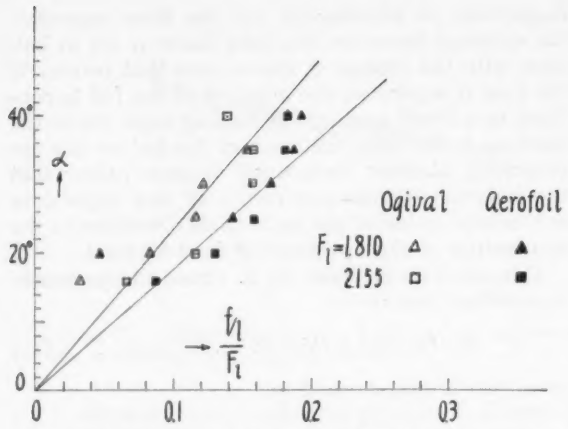

Figure 23. Curves of the criterion for the occurrence of the ventilating flow.

ciding the occurrence of ventilating flow; for a given condition of operation we can plot a point on Figure 23 and then we can decide for the occurrence of the ventilating flow by whether this point is in the upper or lower side of this mean straight line, that is: after the occurrence of ventilating flow or before it.

Further, we can also see from Figure 23 that ventilating flow too occurs less easily in the ogival section than in the aerofoil section since the line for the ogival section is relatively in the upper side.

CONCLUSIONS

In this paper, based on observations and measured pressure distribution, the air drawing and ventilating flow phenomena of a shallowly submerged hydrofoil section may be clarified essentially. The main problems considered may be summarized as follows:

The observed phenomena were described for the inception of air drawing, occurrence of flow separation and form of cavity.

The necessary and adequate condition is considered for occurrence of flow separation due to air drawing, and consequently the surface tension and stall is enumerated.

The effect of the flow separation on the pressure distribution, lift and drag of the hydrofoil section is examined, and also comparison is made with Green's discontinuous flow theory.

The scale effect on the flow separation due to air drawing is examined and the critical Weber number is determined for the NACA 4412 section and for the ogival section.

A criterion is proposed for the occurrence of flow separation and a concrete example is shown.

ACKNOWLEDGMENTS

The author wishes to express his gratitude to Professor F. Numachi and Professor S. Fuchizawa for their support in these investigations.

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LUBRICATES WITH AIR

Use of air as a bearing lubricant is expected to result in wider applications for this type of minimum-friction bearing. Just developed by Boeing is an externally-pressurized step bearing whose load-carrying shaft spins freely on a cushion of compressed air. The Boeing design employs a single opening for the pressurized air.

—THE IRON AGE, May 25, 1961

EDWARD F. NOONAN

PROPELLER SHAFT BENDING STRESSES ON THE S. S. ESSO JAMESTOWN

THE AUTHOR

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FOREWORD

This report written by Edward F. Noonan was prepared by Noonan, Knopfle and Feldman, Professional Engineers, under contract to the Society of Naval Architects and Marine Engineers. The study was recommended by the M-8 Panel (Investigation of Tailshaft Failures). The interpretation, conclusions and opinions are solely those of the author.

SECTION I. BACKGROUND & PURPOSE

As part of the research program on the "Investigation of Tailshaft Failures" the Society of Naval Architects and Marine Engineers sponsored the study of propeller shaft bending stresses on the *S. S. Esso Jamestown*, Newport News Shipbuilding & Dry Dock Co. Hull No. 527. The test installation

was accomplished during the construction of the vessel and was reported on by [1].

Following completion of the test installation data were taken during a series of three trips. Information was obtained under various sea conditions, at various loadings and speeds. This report covers the analysis and interpretation of the data collected during these trips.

The scope of this assignment was given as follows:

I. Scan all test data obtained, classify test conditions, identify runs and select suitable sections of records for analysis.

II. Evaluate all runs for overall stress levels and faired first orders.

III. Perform harmonic analysis on selected runs.

IV. Interpret results and compare with results of previous studies.

V. Prepare report for distribution to all M-8 panel members.

The work presented in this report, when taken with the report of "Strain Gage Installation," [1] is intended to be a complete study in itself. It should be borne in mind however, that this study is only one of a series sponsored by the S.N.A.M.E., The Bethlehem Steel Co., Shipbuilding Division, Quincy, Mass., and the U.S. Navy, Bureau of Ships. The vessels on which such studies have been conducted include:

1. U.S.N.S. *Mission San Luis Obispo,* T-2 Tanker.
2. S.S. *Chryssi,* 29,000 ton Tanker.
3. U.S.N.S. *Lt. James E. Robinson,* Victory Ship.
4. S.S. *Esso Jamestown,* 50,000 ton Tanker.
5. U.S.S. *Norfolk,* Destroyer Leader. (DL-I)

The first four vessels are all single screw vessels and results of the studies have been documented by references [2] thru [6]. This report gives the results of the work on the S.S. *Esso Jamestown.* The results of the study on the U.S.S. *Norfolk* are presently being compiled by the David Taylor Model Basin and should be issued shortly, [7].

The Introduction to the "Strain Gage Installation for the Investigation of Tailshaft Bending Stresses on the S.S. *Esso Jamestown,*" [1] gives a brief background on the importance of bending stresses experienced in propeller shafts. It will suffice at this point to say that the alternating bending stresses are by far the largest alternating stresses experienced in service and as such, are considered to be the largest single factor contributing to the many failures encountered to date. It is by such full scale studies as this that the M-8 Panel hopes to better understand the problem and by sufficient empirical data, form the basis for an improved design procedure.

After completion of the strain gage installation, limited data were obtained during the Builder's Trial, 17 Dec. 1957. The remainder of the data were obtained during the first three service trips of the *Esso Jamestown,* 20 Dec. 1957 thru 22 Jan. 1958.

The data was obtained by the following:
Dr. E. G. Baker, American Bureau of Shipping.
Lt. Commander C. Baker, U.S.C.G.
Mr. Robert Fremlin, Technical Asst., S.N.A.M.E.
The assistance of these men did much to insure the success of this study.

SECTION II. INSTRUMENTATION & CALIBRATION

A—INSTRUMENTATION

A schematic diagram of the strain gage installation is given in Figure 1. This diagram shows the following:

1. The angular position of the strain gages on the tail shaft. These gages are located at the forward end of the tailshaft taper, just forward of the propeller hub.

2. The grouping of the gages to form the strain gage bridges nos. I, II, III, & IV.

3. The method of accomplishing this grouping by the interconnection of the 32 conductors contained in the 4 shaft cables utilizing the terminal strips mounted on a wooden panel secured to the rotating shaft.

4. The method of applying the four terminals of *either* Bridge I or Bridge III to slip rings A, B. C, & D utilizing the shaft switch S1 mounted on the wooden panel. Also, the method of applying the four terminals of *either* Bridge II or Bridge IV to slip rings E, F, G, & H utilizing switch S2.

5. The method of energizing the two selected bridges by applying a constant D.C. voltage (30 V) across brushes A-B and E-F utilizing the heavy duty dry batteries, the controlling series rheostats, and voltmeter.

6. The method of recording the output of the selected bridges by connecting brushes C-D to Edin Amplifier No. 1 and brushes G-H to Edin Amplifier No. 2.

A complete description of the modifications to the shaft, physical location of the gages and method of installation is given in [1]. A detailed description of the operating instructions for the test equipment is given as Appendix A to [1].

An auxiliary pen, actuated by a special ninth slip ring, was provided in the recording oscillograph to impart a revolution mark on the recorded oscillograms. Heat lamps, directed at the slip rings were provided to reduce condensation which could produce "shorts" in the system.

B—EVALUATION OF STRAIN GAGE OSCILLOGRAMS

The basic expression for the output voltage of a D.C. excited strain gage bridge composed of four active arms oriented to respond to bending strain is:

$$mv = 1000\,VK\,\frac{S}{E}$$

where
mv= bridge output signal in millivolts (peak)

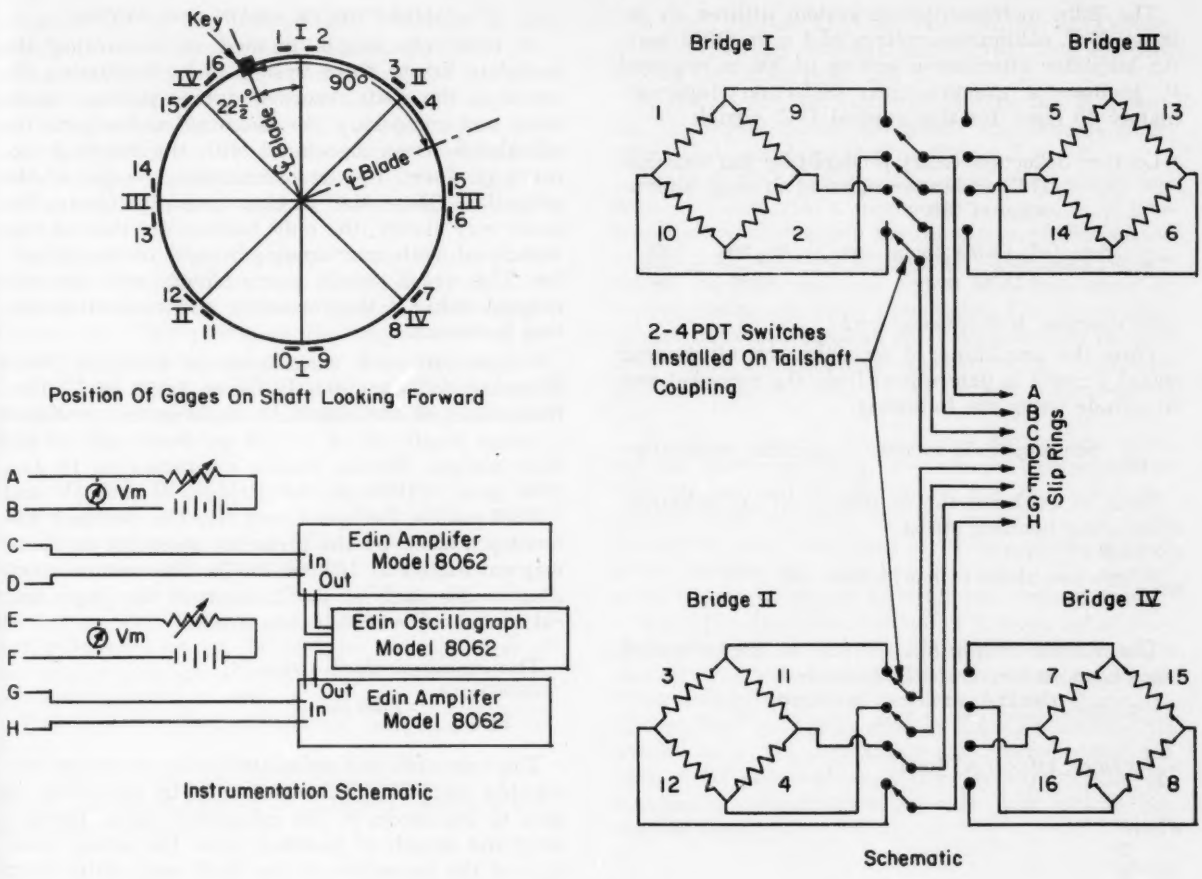


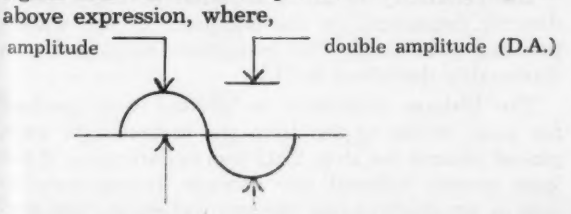
Figure 1. Strain Gage Wiring Diagram.

V=D.C. voltage applied to bridge=30 volts
K=gage factor=3.24 for the CD-7 gages utilized
S=bending stress in psi (peak)
E=modulus of elasticity=30×10^6

$$mv=\frac{1000\times30\times3.24}{30\times10^6}\times S=\frac{3.24}{1000}\times S$$

$$S=\frac{1000}{3.24}\times mv=308\times mv$$

Thus for a condition of alternating bending stress, the amplitude of the instantaneous bridge output signal and the stress amplitude are related by the above expression, where,



The symbol ± (plus or minus) is used to indicate amplitude.

In order to evaluate the bridge output millivolts, and hence the indicated stress, from the recorded oscillogram, the following term is defined:

$$\text{Instrumentation Sensitivity } (I_s)=\frac{\text{pen deflection (mm)}\times\text{amplifier atten.}}{\text{mv applied to amplifier}}$$

The above expression applies to either D.C. or A.C. calibration of the Edin recording instrumentation system utilized. That is, if a *known* D.C. signal were applied:

$$(I_s)=\frac{\text{pen deflection (mm)}\times\text{amplifier atten.}}{\text{D.C. mv}}$$

In like manner, if a *known* A.C. calibration signal were applied (at a frequency within the flat response range of the system) the same sensitivity value (I_s) would be obtained, where,

$$I_s=\frac{\text{pen amplitude } (\pm\text{mm})\times\text{ampl. att.}}{\pm\text{mv}}$$

The Edin instrumentation system utilizes an internal D.C. calibration voltage of 1 volt (1000 mv). An amplifier attenuation setting of 200 is required to produce a practical pen deflection (approximately 10 mm) for this applied D.C. signal:

Let D= deflection (mm) produced by the 1000 mm D.C. calibration signal at an ampl. attenuation of 200.

$$\therefore I_s = \frac{\text{pen defl. (mm)} \times \text{ampl. att.}}{\text{D. C. mv}} = \frac{D \times 200}{1000} = \frac{D}{5}$$

for example, if D=10mm, I_s=2

Thus the amplitude of an unknown alternating signal (±mv) is determined from the recorded pen amplitude using the following:

$$\pm mv = \frac{\text{pen amplitude } (\pm mm) \times \text{ amplifier attenuation}}{I_s}$$

Thus to evaluate the amplitude of an unknown alternating bending stress:

$$\pm S = 308 \times (\pm mv) =$$

$$308 \times \frac{\text{pen amplitude } (\pm mm) \times \text{ampl. att.}}{I_s}$$

The double amplitude (D.A.) of the recorded trace is more conveniently measured:

$$\therefore \pm S = 308 \times \frac{\text{pen D.A. (mm)}}{2} \times \frac{\text{ampl. att.}}{I_s} =$$

$$\frac{154}{I_s} \times \text{ampl. att.} \times \text{D.A. (mm)}$$

where

$$I_s = \frac{D}{5}$$

Therefore,

$$\pm S = \frac{154}{D/5} \times \text{ampl. att.} \times \text{D.A. (mm)}$$

$$= K \times \text{ampl. att.} \times \text{D.A. (mm)}$$

The extreme range of the expected calibration deflection values (D) at amplifier attenuation setting of 200 for both amplifiers, various warm-up time etc. is from 9 to 12 mm. For convenience, the following tabulation is given showing the value of the constant (K) in terms of the calibration deflection (D).

D (mm)	9.0	9.5	10.0	10.5	11.0	11.5	12.0
K	86	81	77	73	70	67	64

The actual rpm value may be evaluated from a recorded oscillogram since the chart speed utilized is constant at 25 mm/sec. Thus, if (A) is the length in mm for 10 revolution marks.

$$\text{rpm} = \frac{10}{A/25} \times 60 = \frac{15000}{A}$$

For example, if 10 revolution marks measured 150 mm,

$$A = 150, \quad \text{rpm} = \frac{15000}{A} = 100$$

C—CALIBRATION OF STRAIN GAGE SYSTEM

A relatively simple method of calibrating the complete Strain Gage System is by evaluating the stress in the shaft recorded during jacking operations and comparing the measured value with the calculated stress associated with the bending moment produced by the overhanging weight of the propeller. Since the jacking operation turns the shaft very slowly, the only load on the shaft is that associated with overhanging weight of the propeller. This value should check closely with the calculated value if the recording and evaluation system is correct.

Calibration runs were made at Newport News Shipyard following the Builders Trials of 17 Dec. Evaluation of the resultant oscillograms produced a stress amplitude of ±1140 psi from each of the four bridges. Similar checks conducted on 10 Jan. 1958 gave ±1090 psi for Bridges II and IV and ±1100 psi for Bridges I and III. The bending moment produced by the propeller assembly in water was calculated as 165,000 in. lb. The section modulus of the shaft at the location of the gages was calculated to be 1930 inches cubed.

The resultant shaft stress is:

$$\frac{165{,}000 \times 12}{1930} = 1{,}030 \text{ psi.}$$

The measured and calculated values compare reasonably well. The difference may be attributed in part to the errors in the calculated value, particularly the length of moment arm. The actual position of the propeller on the shaft may differ from the assumed position. The total range in measured values, from a low of 1090 psi to a high of 1140 psi would indicate a mean of 1115 psi and a tolerance of ±25 psi. This would indicate a bridge and recording system accuracy of ±2.2 percent.

It is important to note that the positions selected for the gages with reference to the keyway (22½ degrees to either side) successfully avoided the difficulty encountered in Bridge I in the *Robinson* study reported in [5]. In that case Bridge I read low. It was rationalized that the flaring of the stress lines about the gage at the top of the keyway produced the low reading.

D—WATERPROOFING & SYSTEM MAINTENANCE

The reliability of an installation of this type is directly dependent on the adequacy of the waterproofing procedure. The procedure employed was thoroughly described in [1].

The leakage resistance to ground was checked for each bridge at the time the instruments were placed aboard the ship. This was to determine if the gage system suffered any damage during installation of the shaft or had become wet during the several months the ship was in the water. All gages read 200 megohms or better. After all wiring was

complete up to the instruments, readings of 150 megohms or better were still obtained.

Some difficulty was reported during the test runs which was traced to condensation on the slip-ring and brush assembly during the post test check conducted by NKF. A "short" was also located in D.C. cable number one. After locating and correcting the faults, readings of upward of 50 megohms was obtained on all bridges.

In this installation, the proper maintenance of the equipment installed in the shaft alley, was considered more critical than the integrity of the gages themselves. This points to the importance of experienced test personnel capable of recognizing such faults. Heat lamps on both the slip rings and brush assembly to combat shaft alley humidity are also considered necessary.

SECTION III. TEST RESULTS

Preliminary runs were conducted during Builder's Trials by personnel of NKF. These runs yielded valuable information while maneuvering, during a crash back and during high speed runs. Results of these runs were included in Appendix B of [1] and are included here on Table 1 for completeness of the report and as a rough comparison with the regularly scheduled tests.

The three regularly scheduled tests were as follows:

1. Dec. 20-Dec. 29, Data obtained by Dr. E. G. Baker, American Bureau of Shipping. Data are presented on Tables 2 and 3.

2. Dec. 31-Jan. 11, Data obtained by Lt. Comdr. C. Baker, United States Coast Guard. Data are presented on Tables 4, 5, and 6.

3. Jan. 14-Jan. 22, Data obtained by Mr. R. Fremlin, Society of Naval Architects and Marine Engineers. Data are presented on Tables 7 and 8.

Tables 1 thru 8 give a tabulation of all significant data. For each run a representative section of tape was chosen after which the maximum section of the record was analyzed for overall stress and for the faired value of the first order. The first order of course, being the most significant. For an evaluation of harmonic content, several sections of records have been harmonically analyzed. The harmonic analysis is given later in the report. (See Section IV).

The data given in tables 2 thru 8 have been plotted as shown in Figures 2 thru 11. Because of changes in weather conditions the data given for several trips required separate curves. In all cases the curves show the range of stresses obtained for a given loading and weather condition. The shaded areas represent ranges of maximum overall stress.

Since the data obtained under a given set of conditions have a rather broad spread in most cases, a shaded area was developed which shows the range of maximum overall stress experienced. There are also peak or minimum stresses that fall outside this shaded area. We may safely say however, that we have a 90 per cent or better probability that all maximum overall stresses will fall within the shaded area.

TABLE I

Test Results Obtained During Period 16-17 Dec. 1957
16 Dec. 1957 Dockside
17 Dec. 1957 Underway (Builders Trials)

Run No.	R.P.M.	BRIDGE I AT MAXIMUM SECTION OF RECORD		BRIDGE I AT MINIMUM SECTION OF RECORD		BRIDGE II AT MAXIMUM SECTION OF RECORD		BRIDGE II AT MINIMUM SECTION OF RECORD	
		overall stress ampl. (±psi)	faired 1st ord. stress ampl. (±psi)	overall stress ampl. (±psi)	faired 1st ord. stress ampl. (±psi)	overall stress ampl. (±psi)	faired 1st ord. stress ampl. (±psi)	overall stress ampl. (±psi)	faired 1st ord. stress ampl. (±psi)
16 DEC. 1957—DOCKSIDE OPERATION									
A*	17	1120	—	—	—	1110	—	—	—
B	23	1120	—	—	—	1140	—	—	—
C	59	1320	1090	—	—	1710	1210	—	—
17 DEC. 1957—UNDERWAY (BUILDERS TRIALS)						50,176 TONS DISPL.**			
D*	104.5	2950	2820	2140	1940	2920	2840	1980	1810
E	105	2840	2800	1350	1680	3000	2920	1900	1720
F	105	3080	2940	2030	1960	3100	2920	2060	1900
G	105 rt. rud.	4060	3500	1400	700	3870	3000	1290	860
H*	105 lft. rud.	5950	5600	1050	—	5600	5160	1120	—
I*	105 crash astern	9100	7340	—	—	7740	6450	—	—

* Pertinent sections of these oscillograms are presented in Appendix A.
** Sea Conditions, calm.
Upon return to the shipyard in the early hours of 18 Dec., the signal from all four strain gage bridges was recorded during jacking operations (very slow shaft rotation due to operation of the jacking motor). Evaluation of the resultant oscillograms produced a stress amplitude of ±1140 psi from each of the four bridges.

TABLE II

Voyage I—Trip IA Draft: 18′6″ Forward—26′4″ Aft; 18′ Forward—24′ Aft*
Displacement: 29,000 tons (Est.) Sea Conditions: Calm

Date	Sea	Run	RPM	Channel 1—Bridge I Overall ±psi	Channel 1—Bridge I 1st order ±psi	Channel 2—Bridge II Overall ±psi	Channel 2—Bridge II 1st order ±psi
20 Dec.	Calm	101	Jacking	980	——	1140	——
21 Dec.	"	102	"	1160	——	1120	——
"	"	104	91	1610	1210	2050	1540
"	"	105	100	——	——	2610	2540
22 Dec.	"	106	101	2660	2590	2080	2010
"	"	107	100	——	——	1750	1610
"	"	108	95	——	——	1610	1400
"	"	109	90	——	——	1470	1400
"	"	110	85	——	——	1190	1120
"	"	111	77	1610	1420	1470	1260
"	"	112	67	1060	910	980	810
"	"	113	57	730	580	810	600
"	"	114	50	770	660	880	740
			Bridge III				
Dec. 24	"	241*	99	2920	2630	——	——
"	"	242*	99	——	——	2360	2240
Dec. 25	"	251*	Jacking	1120	——	1140	——

TABLE III

Voyage 1—Trip 1B Draft: 33′2″ Forward—33′2″ Aft
Displacement: 45,000 tons (Est.) Sea Conditions: A—Calm shoal water
(15′ under keel) B—Moderate, confused sea, gentle roll and pitch

Date	Sea	Run	RPM	Channel 1—Bridge III Overall ±psi	Channel 1—Bridge III 1st Order ±psi	Channel 2—Bridge II Overall ±psi	Channel 2—Bridge II 1st Order ±psi
26 Dec.	A	261	98	2340	1810	2950	2820
"	A	262	97	1740	1210	1270	1070
29 Dec.	B	291	105	2680	2610	2420	2340
"	B	292	100	2520	2240	2340	2280
"	B	293	95	2480	2410	2280	2210
"	B	294	88	2010	1980	2210	2080
"	B	295	82	1740	1610	1960	1820
"	B	296	77	1680	1470	1750	1680
"	B	297	72	1400	1260	1610	1540
"	B	298	62	1120	980	1230	1190
"	B	299	52	1230	1020	1230	1090
"	B	2910	37	1190	1050	1260	1050

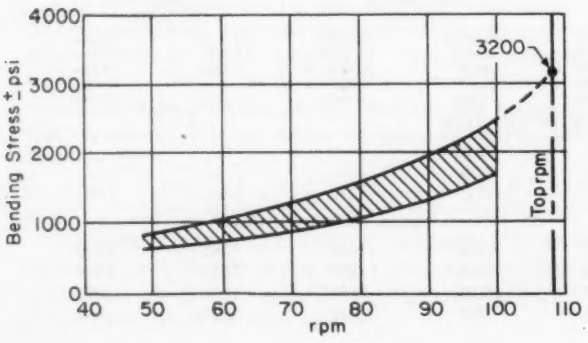


Figure 2.

Voyage 1—Trip 1A
Displacement: 29,000 tons (Est.)
Draft: 18′6″ Forward; 26′4″ Aft
Sea Conditions: Calm
Reference: Table 2

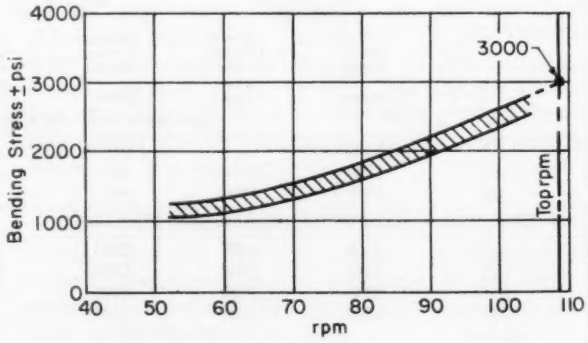


Figure 3.

Voyage 1—Trip 1B
Displacement: 45,000 tons (Est.)
Draft: 33′2″ Forward; 33′2″ Aft
Sea Conditions: Moderate Confused Sea, Gentle Roll and Pitch (3′-5′ Waves)
Reference: Table 3

TABLE IV

Voyage 2—Trip 2A Draft: 14'2" Forward—23'6" Aft
Displacement: 23,500 tons
Sea Conditions: A—Calm shoal water B—Moderate, slight swell, slight roll, no pitching Depth: Depth of water under keel as shown, ft.—d—Deep water, s—Shoal water

					Channel 1—Bridge III		Channel 2—Bridge II	
DATE	SEA	DEPTH	RUN	RPM	Overall ±psi	1st Order ±psi	Overall ±psi	1st Order ±psi
31 Dec.	A	30	2	40	1000	940	800	740
"	A	30	3	78	1410	1210	1340	1000
"	A	30	4	76	1510	1210	1540	1210
"	A	30	5	64	1070	940	1070	800
"	A	30	6	21	1000	1000	1140	1000
"	A	30	7	67	1540	1410	1680	1610
"	A	30	8	78	1270	1070	1210	940
"	B	d	11	95	2800	2660	2180	1920
4 Jan.	A	75	2	66	910	700	870	670
"	A	100	3	88	1750	1540	1740	1470
"	A	100	4A	90	2030	1820	1870	1670
"	A	100	4B	92	1960	1680	1870	1740
"	A	100	4C	92	1750	1470	1740	1470
"	A	100	4D	92	1750	1260	1610	1210
"	A	100	4E	90	1890	980	1810	870
"	A	45	4F	88	1820	1400	1670	1470
"	A	45	4G	86	2100	1680	2140	1870
"	A	45	4H	88	2030	1680	1810	1470
"	A	45	4I	87	2310	2100	2280	2080
"	A	70	7	87	2450	2030	2540	2280
"	A	115	8	88	3290	2590	3020	2680
"	A	50	9	67	840	490	870	540

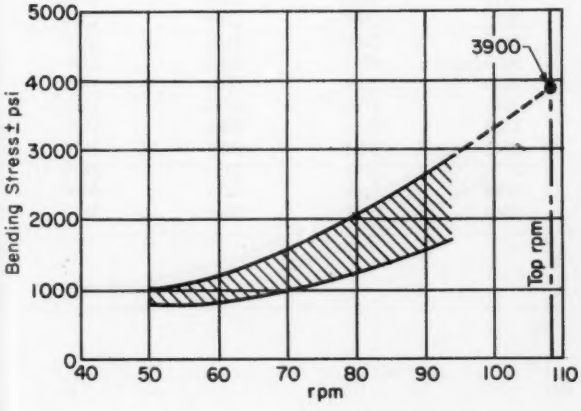


Figure 4.

Voyage 2—Trip 2A
Displacement: 23,500 tons
Draft: 14'2" Forward; 23'6" Aft
Sea Conditions: Calm-*Shoal Water*, Average 30'-100' Under Keel
Reference: Table 4 & 6

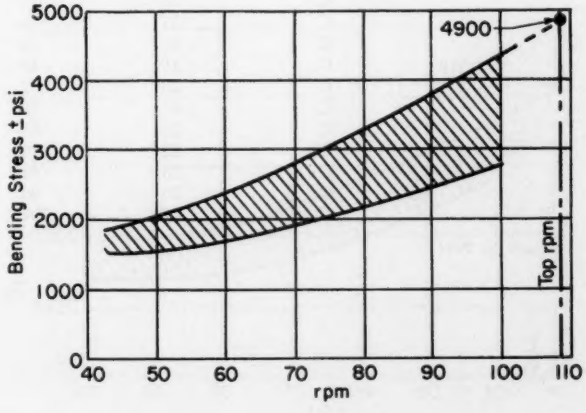


Figure 5.

Voyage 2—Trip 2A
Displacement: 28,700 tons
Draft: 17' Forward; 27' Aft
Sea Conditions: Moderately Rough Following Sea, Slight Roll, No Pitching
Reference: Table 5

TABLE V

Voyage 2—Trip 2A Draft: 17′ Forward—27′ Aft Displacement: 28,700 tons
Sea Condition: A—Moderate, slight roll, very slight pitching B—Moderately rough, following sea, slight roll, no pitching C—Fairly rough (8′-10′ sea), gale force winds, rolling 15° slight pitching

				Channel 1—Bridge III		Channel 2—Bridge II	
DATE	SEA	RUN	RPM	Overall ±psi	1st Order ±psi	Overall ±psi	1st Order ±psi
1 Jan.	A	2	100	2590	2520	2280	2210
"	A	4	99	2940	2800	2680	2540
2 Jan.	B	2	98	2730	2660	2420	2280
"	B	4	99	3500	2520	3750	2820
"	B	5	97	3680	3500	3150	2980
"	B	7	57	1750	1400	1820	1400
"	B	8	21	1260	1220	1260	1220
"	B	9	24	1260	1220	1330	1300
"	B	10	31	—	—	1680	1610
"	B	11	34	—	—	1820	1750
"	B	12	39	—	—	1610	1540
"	B	13	45	—	—	1540	1470
"	B	14	48	—	—	1820	1750
"	B	15	52	—	—	1820	1680
"	B	17	64	3220	2800	2380	2100
"	B	18	70	2800	2450	2450	2280
"	B	19	70	2620	2100	1750	1570
"	B	20	82	4550	3850	3850	3500
"	B	21	82	2980	2100	2450	1750
"	B	22	88	3500	3150	2800	2450
"	B	23	94	3150	2450	2280	2100
"	B	24	100	4340	3220	4340	3080
"	B	25	96	2800	2450	2800	2620
3 Jan.	C	2A	97	3680	3500	3350	3180
"	C	2B	98	4550	3500	4520	3680
"	C	4	95	—	—	3520	3020
"	C	5	87	3000	2800	2620	2540
"	C	6	82	2520	2240	1880	1740
"	C	7	79	3360	2800	3280	2410
3 Jan.	C	8	66	3000	2520	2410	2080
"	C	9	58	2590	2170	2280	1810
"	C	10	50	2240	1960	1940	1740
"	C	12	40	1680	1540	1340	1210
"	C	13	29	1400	1400	1340	1340
"	C	14	18	1260	1260	1210	1210
"	C	16	18	1470	1400	1510	1470
"	C	17	31	2170	1960	2000	1870
"	C	18	41	2520	2380	2410	2210
"	C	19	49	2800	2450	2340	2160
"	C	20	57	2450	2100	1670	1670
"	C	21	59	2660	2170	2210	1870
"	C	22	69	2380	2240	2080	1870
"	C	23	78	2240	1820	1670	1610
"	C	25	92	2730	2170	1940	1870
"	C	26	96	3720	3360	3210	3080
"	C	27*	100	3360	2380	2810	1810

* Turn to Port.

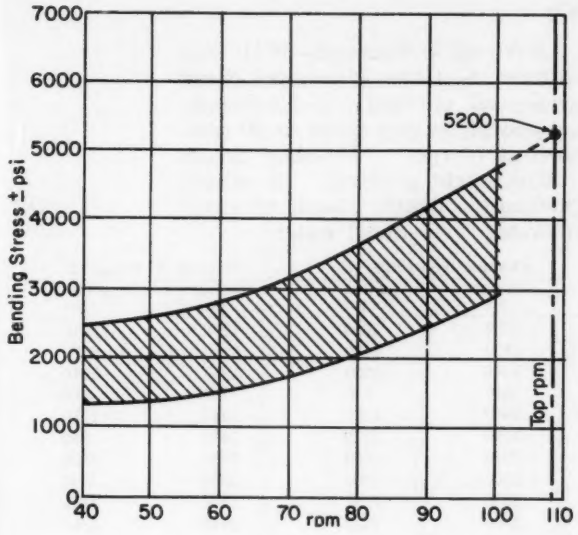


Figure 6.

Voyage 2—Trip 2A
Displacement: 28,700 tons
Draft: 17′ Forward; 27′ Aft
Sea Conditions: Fairly Rough, 8′-10′ Waves, Gale Force Winds, Rolling 15 deg., Slight Pitching
Reference: Table 5

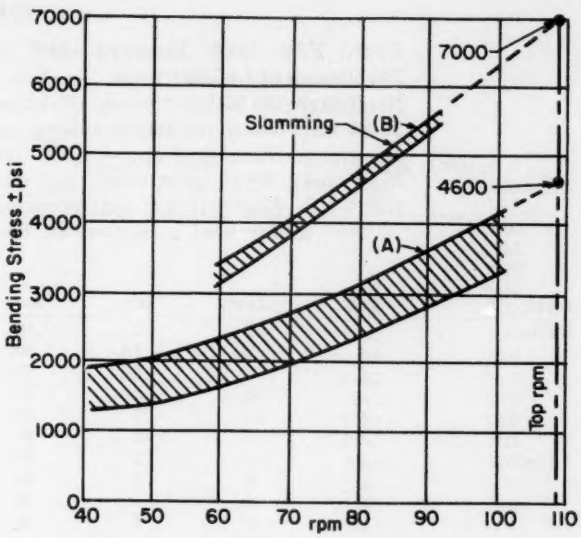


Figure 7.

Voyage 2—Trip 2B
Displacement: 46,000 tons
Draft: 33′7″ Forward; 33′11″ Aft
Sea Conditions: (A) Rough, 30-35 knot Wind, 10′-12′ Waves, Gentle Roll, Pitching. (B) Pitching and Slamming, Slamming Appreciably Diminished at 70 rpm
Reference: Table 6

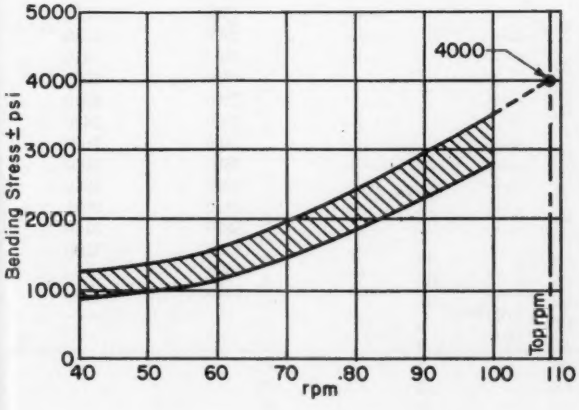


Figure 8.

Voyage 2—Trip 2B
Displacement: 46,000 tons
Draft: 33′7″ Forward; 33′11″ Aft
Sea Conditions: Short Rough Sea, 8′-10′ Waves, 30-35 knot Wind, Not Much Rolling, Slight Pitching
Reference: Table 6

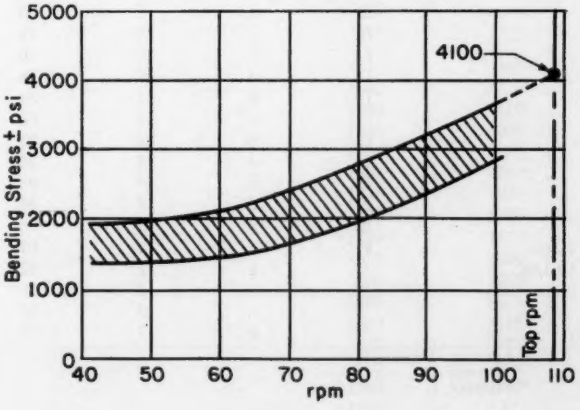


Figure 9.

Voyage 3—Trip 3A
Displacement: 28,000 tons
Draft: 18′9″ Forward; 25′3″ Aft
Sea Conditions: Confused Sea, 5′-8′ Waves, Slight Roll and Pitch
Reference: Table 7

TABLE VI

Draft: F.W.—34′5″ Forward—34′9″ Aft S.W.—33′7″ Forward—33′11″ Aft
Displacement: 46,000 tons Sea Conditions: A—Calm, Mississippi River B—Rough, 30-35 knot wind, 10′-12′ sea, gentle roll, pitching C—Rough, 30-35 knot wind, moderate rolling, pitching enough to ship water at 90 rpm, slamming. Slamming appreciably diminished at 70 rpm D—Short rough 8′-10′ seas, 30-35 knot wind, not much rolling, slight pitching E—Slight roll F—Short rough sea, occasional pitching Depth: Depth of water under keel as shown, ft. d—Deep water s—Shoal water

					Channel 1—Bridge III		Channel 2—Bridge II	
DATE	SEA	DEPTH	RUN	RPM	Overall ±psi	1st Order ±psi	Overall ±psi	1st Order ±psi
6 Jan.	A	75	2	76	1330	1050	1270	1010
"	A	75	3A	80	1400	1120	1210	1070
"	A	75	3B	77	1540	1330	1270	1210
"	A	55	5	57	840	700	740	670
"	A	14	7	79	1680	1260	1940	1410
"	A	12	8	82	1680	1120	1410	940
7 Jan.	B	d	2	99	3850	3680	4020	3350
"	B	d	3	89	3500	2800	3350	2340
"	B	d	4	78	2980	2280	2340	2180
"	B	d	5	66	2800	2380	2410	2280
"	B	d	6	57	2310	2030	1880	1610
"	B	d	7	48	2030	1820	1480	1410
"	B	d	8	42	1960	1680	1340	1270
"	B	d	9	28	1540	1470	1480	1410
"	B	d	10	25	1540	1400	1210	1210
"	B	d	12	23	1610	1540	1340	1340
"	B	d	13	34	—	—	1270	1210
"	B	d	14	45	—	—	1480	1410
"	B	d	15	52	—	—	2080	1940
"	B	d	16	64	—	—	1740	1680
"	B	d	17	72	—	—	2010	1880
"	B	d	18	84	—	—	2510	2340
"	B	d	19	92	—	—	3020	2850
"	B	d	20	100	—	—	3350	3180
"	C	d	23A	92	3150	2800	3680	3350
"	C	d	23B	90	5430	4900	5530	4700
"	C	d	26	64	3680	3150	3350	3180
8 Jan.	D	d	2	74	—	—	1610	1340
"	D	d	3	60	—	—	1140	810
"	D	d	4	40	—	—	1270	1140
"	D	d	5	38	—	—	1140	1070
"	D	d	6	30	—	—	1270	1270
"	D	d	8	48	—	—	1880	1740
"	D	d	9	70	—	—	1810	1610
"	D	d	10	78	—	—	1880	1680
"	D	d	11	84	—	—	2280	2010
"	D	d	12	88	—	—	2620	2480
"	D	d	13	93	—	—	2680	1000
"	D	d	14	98	—	—	2680	1000
"	E	d	16	98	—	—	3020	2850
"	F	d	18	98	—	—	3180	3020
"	E	d	19	96	—	—	3520	3180
9 Jan.	F	d	2	97	—	—	3350	3180
10 Jan.	*	s	2	12	—	—	1090	—
"	"	s	3	12	—	—	1090**	—
"	"	s	6	12	1100***	—	—	—
"	"	s	7	12	1100	—	—	—

* Steam Jacking.
** Bridge IV.
*** Bridge I.

TABLE VII

Voyage 3—Trip 3A Draft: 18'9" Forward—25'3" Aft
Displacement: 28,000 tons Sea Conditions: A—Confused, waves 6'-8', slight roll and pitch B—Confused sea, waves 5'-6', slight roll and pitch
C—Moderately rough sea, slight roll and pitch

DATE	SEA	RUN	RPM	Channel 1—Bridge		Channel 2—Bridge II	
				Overall ±psi	1st Order ±psi	Overall ±psi	1st Order ±psi
14 Jan.	A	1	100	—	—	3280	3080
"	A	2A	99	—	—	3850	3680
"	A	2B	99	—	—	3080	2820
"	B	3	97	—	—	3350	2950
"	B	4	93	—	—	3020	2820
"	B	5	86	—	—	2750	2540
"	B	6	82	—	—	2480	2280
"	B	7	74	—	—	2540	2010
"	B	8	71	—	—	2010	1740
"	B	9	67	—	—	1610	1200
"	B	10	62	—	—	2540	2010
"	B	11	57	—	—	1840	1470
"	B	12	52	—	—	2010	1670
"	B	13	48	—	—	1880	1540
"	B	14	43	—	—	1740	1470
"	B	15	40	—	—	1740	1540
"	B	16	31	—	—	1670	1610
"	B	17	19	—	—	1170	1170
15 Jan.	C	1	103	—	—	2950	2680
"	C	2	87	—	—	2280	2010
"	C	3	76	—	—	1740	1540
"	C	4	67	—	—	1570	1340
"	C	5	57	—	—	1410	1000
"	C	6	42	—	—	1470	1200

TABLE VIII

Voyage 3—Trip 3B Draft: 34'8" Forward—34'8" Aft
Displacement: 46,300 tons Sea Conditions: A—Calm, shoal water (average 5' under keel) B—Mild sea, 3'-5' waves, gentle roll and pitch
C—Ground swells, 5'-10' waves, slight pitch & roll

DATE	SEA	RUN	RPM	Channel 1— Bridge		Channel 2—Bridge II	
				Overall ±psi	1st Order ±psi	Overall ±psi	1st Order ±psi
18 Jan.	A	1	76	—	—	1410	960
"	A	2	42	—	—	1540	1410
"	A	3	68	—	—	1730	1310
"	A	4	76	—	—	1540	1020
"	A	5	44	—	—	1600	1410
"	A	6A	75	—	—	1920	1340
"	A	6B	77	—	—	1470	1020
"	A	7A	58	—	—	1500	1220
"	A	7B	60	—	—	1790	1410
"	A	8A	80	—	—	2820	2560
"	A	8B	80	—	—	2750	1280
"	A	8C	80	—	—	3900	3070
"	A	9	102	—	—	2430	1410
21 Jan.	B	1	104	—	—	2430	2180
"	B	2	90	—	—	1980	1660
"	B	3	80	—	—	1660	1340
"	B	4	70	—	—	1220	1020
"	B	5	56	—	—	1090	900
"	B	6	50	—	—	960	770
"	B	7	40	—	—	960	830
"	B	8	104	—	—	2820	2430
22 Jan.	C	1	60	—	—	1060	960
"	C	2	40	—	—	1090	900
"	C	4	76	—		1660	1410

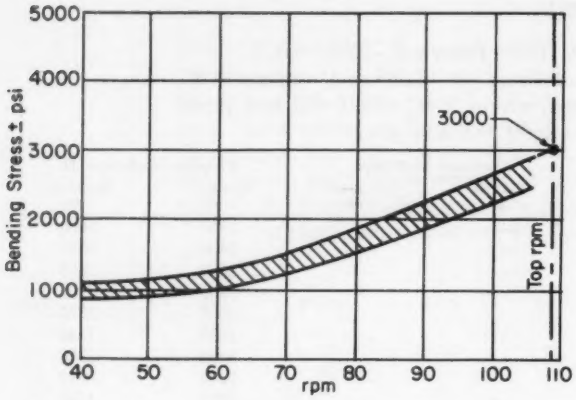


Figure 10.

Voyage 3—Trip 3B
Displacement: 46,300 tons
Draft: 34'8" Forward; 34'8" Aft
Sea Conditions: Moderate Sea, 3'-5' Waves, Slight Roll and Pitch
Reference: Table 8

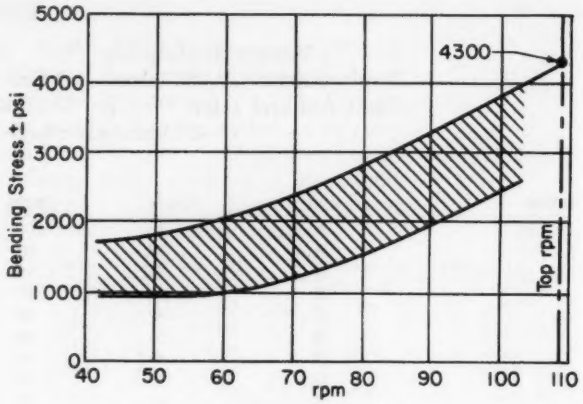


Figure 11.

Voyage 3—Trip 3B
Displacement: 46,300 tons
Draft: 34'8" Forward; 34'8" Aft
Sea Conditions: Calm, *Shoal Water*, Average 5' Under Keel
Reference: Table 8

For comparison purposes, we have extrapolated the upper curve of the envelope to obtain a theoretical maximum overall stress anticipated at the maximum design rpm of 108.5. These values are tabulated in the summary sheet, table 9 for convenience. Also included in the Summary Sheet are the important stress values obtained during the Builder's Trials.

The data given in the Summary Sheet has been arranged in groups for convenience. The first group, lines 1 thru 5 gives the results of full load measurements made during sea conditions of increasing severity. We observe that the anticipated stress for the moderate seas given in lines 1 and 2 agree well with the value obtained during the Builder's Trials at about 3,000 psi. As the sea conditions grow more severe, the bending stresses increase to a maximum of about 7,000 psi. The range for loaded condition, during various sea states, may be given as 3,000 psi for calm seas to 7,000 psi for rough seas.

Lines 6 thru 9 on the summary sheet gives similar values for the light load condition. The stress range anticipated is 3,200 psi for calm conditions and 5,200 psi for fairly rough seas. No measurements were made during extremely rough conditions similar to those referred to in line 5. The maximum value of 5,200 psi given in line 9, may be considered comparable to the value of 4,600 psi given in line 4, for the loaded condition.

Lines 10 and 11 represent shoal water operations. Most of these measurements were made on the Mississippi River and of course, represent calm sea conditions. We note in line 10 of the Summary Sheet, that a maximum value of 3,900 psi may be obtained vs. 3,200 psi for similar loading and similar sea condition, in deep water. Hence, we may conclude that increased stresses may be expected in shallow water.

Line 11 gives a value of 4,300 psi for full load, calm seas and very shallow water. This value is substantially greater than the value of 3000 psi observed for deep water operation, lines 1, 2 and 12. From these data also, we may conclude that bending stresses may be expected to increase as we go from deep to shallow water.

As a further substantiation of this fact we may note, that while light ship conditions resulted in increased bending stress over heavy ship conditions for equivalent sea states, (3,200 psi to 5,200 psi for light condition vs. 3,000 psi to 4,600 psi for heavy condition), line 11 gave a higher stress for the heavy condition than did line 10 for the light condition, (4,300 psi vs 3,900 psi). This may be explained by noting the depth of water. An average depth of only 5 feet under the keel is given on line 11, while varying depths of 30'-100' under the keel are given on line 10. Thus, the effect of depth of water under the keel was found to be of sufficient importance for shoal water operation as to overcome the importance of displacement.

The effects of maneuvering may only be ascertained from the data obtained on the Builders trials. In this respect we may observe that stresses as much as twice the normal operating stress, may be expected during hard turns. Stresses as high as three times the normal operating stress may be encountered during a crash back, from full ahead to full astern.

Our observations may be summed up by using a set of factors for estimating stresses *on this vessel* at any particular operating condition, using full load, calm seas, deep water and straight ahead operation as a base line. This set of factors applies to rated speed and is given as follows:

Load	Sea Condition	Operation	Depth	Factor
Heavy	Calm	Ahead	Deep	1
"	"	Maneuvering	"	2
"	"	Crash Back	"	3
Light	"	Ahead	"	1⅛
"	"	Maneuvering	"	2¼
"	"	Crash Back	"	3⅜
Heavy	Stormy	Ahead	"	2½
Light	"	"	"	2⅔
Heavy	Calm	"	Shoal	1½
Light	"	"	"	1¾

To find the anticipated maximum bending stress for any operating condition for this vessel, multiply the full load operating stress of 3,000 psi by the appropriate factor given above. Since the crash back operation is a very rare occurrence, it is suggested that for design purposes, a maximum factor of 2⅔ or 3 be used in estimating maximum anticipated operating bending stresses.

Referring back to the calculated bending stress of 1040 psi due to the overhanging weight of the propeller we note that this value is roughly one third of our reference stress of 3,000 psi. Thus we see that in this particular case, the maximum anticipated bending stress in service is estimated at 3×3 or nine times the calculated bending stress due to the propeller overhang.

SECTION IV. HARMONIC ANALYSIS

The bending stresses measured on the *Esso Jamestown* were similar in wave form to those measured on the previous tests of this type [2, 3, 5]. In all previous cases the oscillograph record of the bending stress was a complex wave which, when harmonically analyzed, was found to include the following principal components:

(a) First order stresses, corresponding to the frequency of shaft rotation:
(b) Counterwhirl stresses, corresponding to the frequency of the propeller blades minus the frequency of shaft rotation;
(c) Forward whirl stresses, corresponding to the frequency of the propeller blades plus the frequency of shaft rotation;
(d) Higher harmonics of both the counterwhirl and forward whirl frequencies.

By scanning the previous reports [2, 3, 5], we note that the 1st order stress is by far the most significant and appears to have an average value of 75 per cent to 95 per cent of the overall stress. There is however, considerable divergence from this value in some cases, particularly when the wave form is varying appreciably under rough

TABLE IX

Maximum Anticipated Stress at Rated R.P.M.

Line	From Fig.	Displ. Tons	Draft Fwd.—Aft	Sea Condition	Max. Stress ±psi
1	3	45,000	33'2"—33'2"	Moderate confused sea, 3'-5' waves, slight roll & pitch.	3,000
2	10	46,300	34'8"—34'8"	Moderate sea, slight roll & pitch.	3,000
3	8	46,000	33'7"—33'11"	Short rough sea, 8'-10' waves, 30-35 knot wind, not much rolling, slight pitching.	4,000
4	7	46,000	33'7"—33'11"	Rough sea, 10'-12' waves, 30-35 knot wind, gentle roll, some pitching.	4,600
5	7	46,000	33'7"—33'11"	Rough sea, 10'-12' waves, 30-35 knot wind pitching heavily, pounding.	7,000
6	2	29,000	18'6"—26'4"	Calm.	3,200
7	9	28,000	18'9"—25'3"	Confused sea, 5'-8' waves, slight roll & pitch.	4,100
8	5	28,700	17'—27'	Moderately rough, following sea, slight roll, no pitching.	4,900
9	6	28,700	17'—27'	Fairly rough, 8'-10' waves, gale force winds, rolling 15°, slight pitching.	5,200
10	4	23,500	14'2"—23'6"	Calm, shoal water, 30'-100' depth under keel.	3,900
11	11	46,300	34'8"—34'8"	Calm, shoal water, 5' depth under keel.	4,300
			Builders Trials		
12	—	50,176	36'9"—36'9"	Calm—ahead at 105 rpm.	3,100
13	—	50,176	36'9"—36'9"	Calm—right rudder, 105 rpm.	4,000
14	—	50,176	36'9"—36'9"	Calm—left rudder, 105 rpm.	5,900
15	—	50,176	36'9"—36'9"	Calm—crash astern from full ahead.	9,000

sea conditions. In such cases, sections of records may be obtained where the range may vary from slightly below 50 per cent to slightly above 100 per cent of the overall stress.

Second in importance has been the stresses at counterwhirl frequency which appear to average at about 15 per cent of the overall stress. Here too, however, variations could be found which would occasionally reach 20 to 25 per cent.

Stresses at forward whirl frequency generally averaged slightly less than those at counterwhirl frequency and rarely exceeded a value of 20 per cent. The average may be estimated at about 12 to 15 per cent of the overall stress.

An exception to the above pattern was observed in the case of the SS *Chryssi* investigation, when the vessel was equipped with a 5 bladed propeller. Referring to Table 2 of [3], the average 1st order stress was 45 per cent of the overall, the average 4th order stress was 35 per cent of the overall and the average 6th order was 32 per cent of the overall.

Under approximately the same conditions of operation the following proportions were noted when the *Chryssi* was fitted with a 6 bladed propeller; 1st order, 72 per cent; 5th order, 14 per cent; and 7th order, 19 per cent. These averages were taken from Table 9 of [3]. In this particular selection of runs, we note that the 7th order value, exceeds the 5th order. Considering all data, on all ships tested, the stresses associated with the forward whirl frequency appear to be equal to or slightly less than the stresses associated with the counterwhirl frequencies.

As a check on the make-up of the complex waves recorded on the *Esso Jamestown,* typical sections of records were harmonically analyzed for typical operating conditions of light and heavy loading, calm and rough seas, deep and shoal water. Results of these analyses are given in Table 10.

The results of the analyses shows that the relative importance of the orders analyzed, follow the general pattern of other ships tested, when equipped with propellers having an even number of blades. An exception to this trend is noted in the case of run 8B during which time only 5 feet of water was reported under the keel. The ground effect apparently greatly increased the relative importance of the 3rd order, the fundamental counterwhirl frequency.

A study of the harmonic content of the stress pattern was made in reference 4, for the calculated stress pattern It was noted that the content was principally composed of 1st order, n−1 order, n+1 order where n= the number of propeller blades. A small percentage of 2n−1 and 2n+1 orders was also present. That these orders essentially comprised the total content was clearly shown by taking the analyzed data and synthesizing the complex wave. Very good agreement was noted.

In this study, an attempt was made to synthesize the actual oscillograph record from the harmonic analysis. Figure 12 shows the result of this synthesis. Again, good agreement occurs.

The significant observations to be made in regard to the complex wave forms of the actual oscillograph records are that the 1st order stress is by far the largest single component and that although the higher orders may be significant in certain cases, the average stress value may be adequately described by the overall stress value. Since, in the normal experience of an operating vessel, a sufficient number of cycles of reverse bending are expected to occur to permit a fatigue failure at a first order frequency, there is little reason to give great concern to the higher orders. In lieu of attempting to design for particular orders, the use of the overall stress value would appear to be an adequate method of accounting for the presence of the higher harmonics. This simplified approach is further justified by the wide variations in stress levels we may experience on the rough sea condition, when lightly loaded, the combination, which is one of the more severe conditions encountered in normal operation.

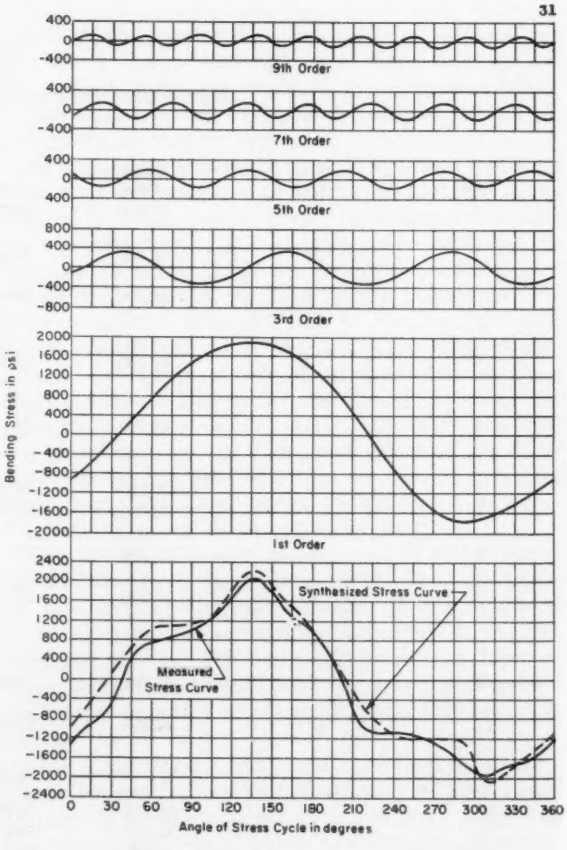


Figure 12. Harmonic Analysis of Run 292 Heavy Load-Calm Sea.

TABLE X

HARMONIC ANALYSIS SUMMARY

Run No.	Condition	Overall ±psi*	1st Order ±psi	%	3rd Order ±psi	%	5th Order ±psi	%	7th Order ±psi	%	9th Order ±psi	%
105	Light Load calm sea	2500	2600	103	160	6.5	160	6.5	108	4.3	95	3.8
2B	Light Load rough sea	4850	4050	83.5	635	13.1	328	6.8	70	1.4	117	2.4
292	Heavy Load calm sea	2070	1800	87.0	315	15.2	125	6.1	110	5.3	63	3.0
23B	Heavy Load rough sea	5700	5230	91.7	214	3.8	402	7.1	100	1.8	47	0.8
7	Light Load shoal water	2600	2100	80.7	181	7.0	278	10.6	224	8.6	202	7.8
8B	Heavy Load shoal water	2940	2400	81.7	1230	41.7	276	9.4	18	0.6	negl.	

* The section of record chosen for analysis was considered typical in wave form for that run. The overall stress value does not necessarily correspond to the maximum value for that particular run.

SECTION V. PHASE RELATIONSHIP

It was shown in [5] that the overall bending stress is dependent on three principal components of bending moment, the thrust moment, the torque moment and the gravity moment. It was also shown in [5] that the calculated values of these moments indicated that the thrust moment was significantly greater than the others. The gravity moment was in turn, significantly greater than the torque moment.

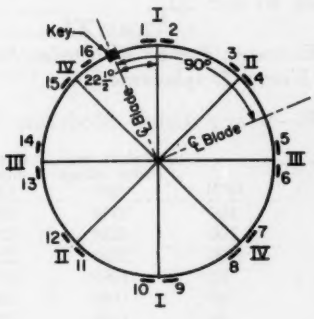
The combined stress pattern is the resultant stress produced by the above moments, and their harmonics. It was shown earlier, and in the results of previous tests, that the harmonic content of the combined stress pattern was of relatively small importance. It may be said then, that the first order stress roughly represents the stress produced by the combined moments referred to above and that its magnitude is a direct function of the phase relationship of the various vectors.

The phase relationship between the gravity moment and the combined first order moment was studied for comparison with the *Robinson* test and to estimate its influence on the stress pattern. To facilitate this study, a "top-center" revolution marker was installed on the shaft. The position of the "top-center" was arbitrarily placed on the shaft. As shown in figure 13, the gage locations are spaced every 45°.

A more exact determination of the location of the revolution marker with respect to the gravity stress was made by recording the bending stresses on all four bridges during the jacking runs of 17 Dec. As seen in Figure 13, the bottom peak of the gravity component leads or lags the "make" point by an angle B_{tc}. The following values of B_{tc} were obtained:

Bridge	B_{tc}
I	126° lag
II	80° lag
III	35° lag
IV	10° lead

Thus, for the section of the oscillograph record chosen to be analyzed, the angle Bg can be determined.



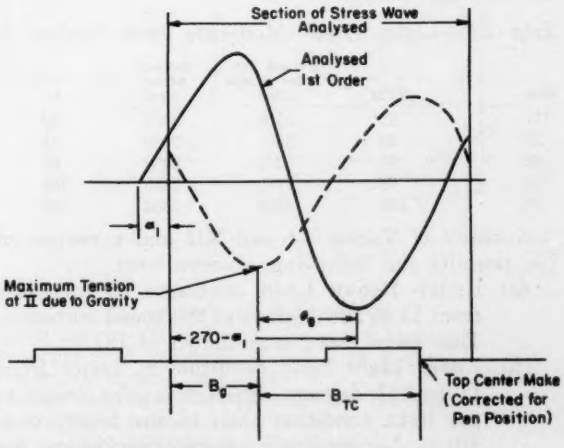


Figure 13. First Order Phase Relationship.

Considering the stress in Bridge II, let

α_1= angle of lead of 1st order component.

B_{tc}= angle between bottom of gravity stress wave and "make."

B_g= angle between arbitrary reference point and bottom of gravity stress wave.

θ_g= phase angle between 1st order and gravity stress component.

θ_g= $270-(\alpha_1+B_g)$.

From Figure 13 we may observe that if B_g is less than $(270-\alpha_1)$, then the 1st order leads the gravity component. If B_g is greater than $(270-\alpha_1)$, then the 1st order lags the gravity component.

The phase relationship between the 1st order stress and the gravity stress was established for various speeds in the heavy and lightly loaded conditions. As near as possible, comparative sea conditions were used. Since few of those runs which were harmonically analyzed were included in the phase relationship study, and since few of the harmonically analyzed records included "top-center" marks, the phasing relationship of the gravity moment was determined with respect to the "faired" 1st order.

Results of the phase relationship study are given in Tables XI and XII:

TABLE XI:

Angle Between Faired 1st Order Stress & Gravity Stress, Heavy Displacement.

Trip 1B—Heavy Load—Moderate Sea—Bridge II

Run	RPM	Faired 1st Order Stress ±psi	Overall Stress ±psi	θ_g°
291	106	2340	2420	130
292	100	2280	2340	110
294	90	2080	2210	101
296	80	1680	1750	89
298	65	1190	1230	50
2910	40	1050	1260	13

TABLE XII:

Angle Between Faired 1st Order stress & Gravity Stress, Light Displacement.

Trip 2A—Light Load—Moderate Sea—Bridge II

Run	RPM	Faired 1st Order Stress ±psi	Overall Stress ±psi	θ_g°
17	65	2100	2380	26
20	80	3500	3850	11
22	90	2450	2800	80
23	95	2100	2280	103
25	100	2620	2800	103

A study of Tables XI and XII and a review of [5] permits the following observations:

(a) Under Heavy Load condition θ_g increases from 13 to 130 degrees as the speed increases. This same trend was shown in [5]

(b) Under Light Load condition θ_g varies from 11 to 103 degrees. Smaller angles occur in the light condition than in the heavy condition. Larger angles were observed on the *Esso Jamestown* than on the *Robinson* for the light condition.

(c) The closer θ_g approaches 180°, the more "out of phase" the 1st order becomes, with respect to the gravity stress. This reduces the combined stress.

(d) Referring to Table XI, we note the phase angle increases with rpm and thrust. As we approach top speed θ_g is 130 degrees and is nearly out of phase. We can roughly deduce, that since the measured 1st order stress is over twice the gravity moment, the eccentric thrust moment must be roughly equal to three times the gravity moment.

(e) Referring to Table XII, we note a trend toward higher angles of θ_g with higher speed, as in the case of the heavy load condition. Average stresses are higher in the light load condition and the angle θ_g is somewhat smaller.

(f) Run No. 20 of Table XII shows a relatively high stress with a corresponding lower angle θ_g. This is probably due to a pitching motion.

The above data confirm in principle the data obtained on the USNS *Lt. James E. Robinson* and definitely establish that the center of thrust is eccentric to the axis of the shaft and produces a bending moment several times greater than that produced by the overhanging weight of the propeller. It is also apparent that under normal immersion of the wheel (heavy load) this moment is partially compensated for, by the gravity moment, indicating the center of thrust is above the axis of the shaft. Under light loads, or rough sea conditions which produce pitching of the vessel, this center of pressure falls below the axis of the shaft and augments the gravity moment. Comparisons of actual values of stress measured on various ships and alternate methods of treating the problem in the design state are discussed in the following sections.

SECTION VI COMPARSION WITH PREVIOUS TESTS

The test program has two fundamental objectives. The first is to obtain a better understanding of the phenomena associated with the shaft failures and the second is to obtain empirical data which would assist in the development of a more reliable design procedure. As previously stated, the test conducted on the SS *Esso Jamestown* is the fourth in a series of tests carried out on single screw vessels. As a rough check on the information obtained to date, a comparison is drawn between the results obtained on all four ships measured.

All tests to-date have shown that the tailshaft bending stresses vary widely over loading conditions, sea conditions, speed, depth of water and maneuvering conditions. It was also apparant that a rather large spread of data was obtained for various runs at a fixed set of operating conditions. An example of this spread is noted in the curves shown in figures 2 to 11.

It was also apparent that the shafts of all ships measured were not designed to the same standards. By that is meant, some shafts were larger than called for by the A. B. S. rules while one was slightly under.

To make what may be called a reasonable comparison between the various designs and associated bending stresses, the following conditions were established:

(a) All stresses were corrected to equivalent stresses obtained with a shaft designed to ABS requirements;

(b) Light and heavy displacements were used for comparison purposes;

(c) Maximum overall stress conditions were used and extrapolated to top rated RPM.

(d) In the case of the SS *Chryssi*, results shown were obtained with a six bladed propeller. It is expected that the measured values would have been somewhat higher if a four bladed propeller had been used.

(e) The most difficult problem in making such a comparison is the estimation of sea conditions. The curves shown in Figures 14 thru 17 have been chosen as being as representative of similiar sea conditions as possible. Calm seas are represented, which may be taken as fair weather with only slight rolling and pitching and waves not exceeding 3 to 5 feet.

In estimating the normal A.B.S. shaft size, the following expressions were used:
where:

$$d = c\sqrt[3]{\frac{K H}{R}}$$

d = shaft diameter in inches (solid)
c = constant (.95 for line shafts)
K = constant (64 for ocean vessels)
H = shaft horsepower
R = revolutions per minute

$$D = 1.05 \times d + \frac{C}{P}$$

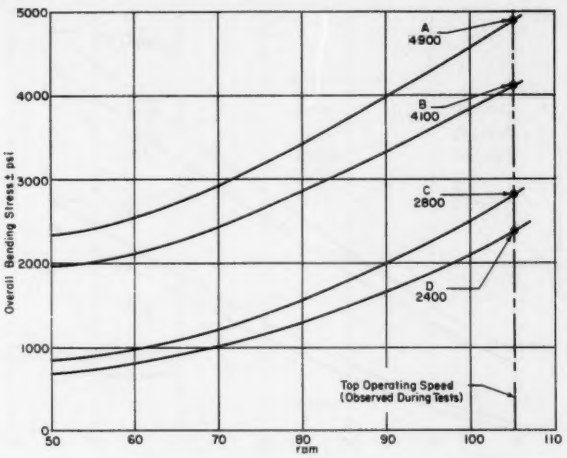
where:
D = Tailshaft diameter in inches (solid)
P = Propeller diameter in feet
C = Constant (12 for continuous liner)

In estimating the normal thrust of the vessel, the following expression was used:

$$T = \frac{326 \times EHP}{V \times (1-t)}$$

where:
T = Thrust in pounds
EHP = SHP × Propulsive Coefficient (P_c)
V = Rated speed in knots
t = Thrust deduction factor

The maximum anticipated stresses for the light and heavy conditions given in figures 14 thru 17



.. Figure 14. U.S.N.S. *Mission San Luis Obispo.*
10,000 shp 95 rpm 15.7 knots 4 blades
Propeller Shaft Diameter: 20⅛" by A. B. S. 20½"
Correction Factors: Size=1.05 Gage Location=0.80
Combined=0.840
Overall Bending Stress ± psi Calm Sea Condition
A—Light Displacement—Measured
B—Light Displacement—Corrected
C—Heavy Displacement—Measured
D—Heavy Displacement—Corrected
Reference: [2] Tables 4 & 5

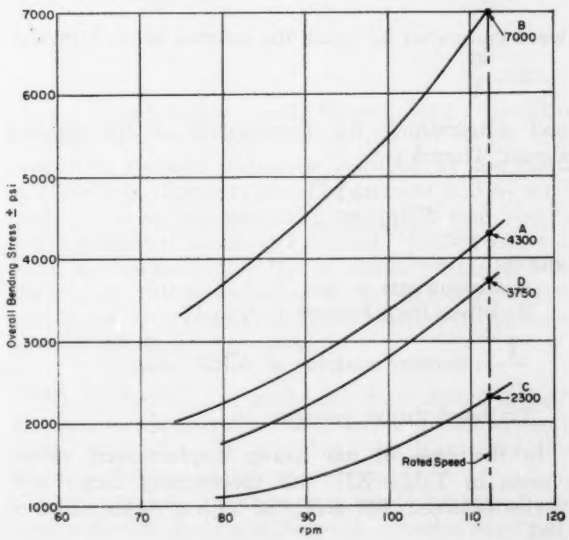


Figure 15. S.S. *Chryssi.*
15,000 shp 112 rpm 17 knots 6 blades
Propeller Shaft Diameter: 26" by A. B. S. 22¼"
Correction Factors: Size=1.64
Overall Bending Stress ± psi Calm Sea Condition
A—Light Displacement—Measured
B—Light Displacement—Corrected
C—Heavy Displacement—Measured
D—Heavy Displacement—Corrected
Reference: [3] Tables 4 & 6

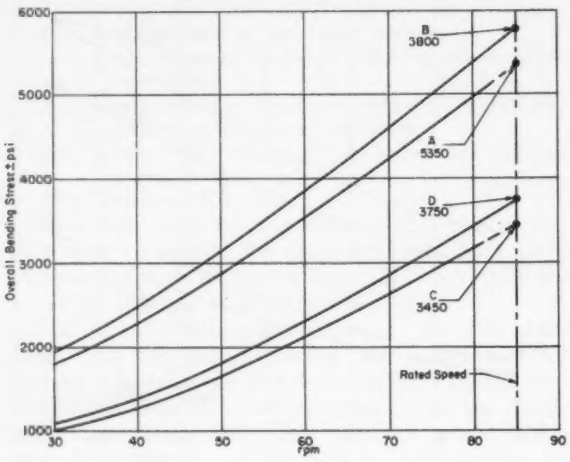


Figure 16. U.S.N.S. *Lt. James E. Robinson.*
8500 shp 85 rpm 17.55 knots 4 blades
Propeller Shaft Diameter: 21″ by A. B. S. 20½″
Correction Factor: Size=1.08
Overall Bending Stress ± psi Calm Sea Condition
A—Light Displacement—Measured
B—Light Displacement—Corrected
C—Heavy Displacement—Measured
D—Heavy Displacement—Corrected
Reference: [6] Table 7

were compared by using the normal beam formula:

$$S=\frac{M_c}{I}$$

and determining the eccentricity of the applied thrust. Therefore:

$$e=\frac{S\times I}{12\times T\times c}$$

where:

e=Eccentricity in feet
S=Stress from Figures 14 thru 17

$\frac{I}{c}$ = Section modulus of A.B.S. shaft

T=Rated thrust, pounds.

In the case of the heavy displacement values shown in Table XII, the eccentricity factor e/P varies between .047 and .082 with a mean value of .061.

In the case of the light displacement values shown in Table XI, the eccentricity factor e/P varies between .075 and .125 with a mean value of .098.

The eccentricity factor for light loads may be taken as 1.6 times the equivalent factor for heavy loads. This value of 1.6, for fair weather, with only slight rolling and pitching and waves not exceeding 3 to 5 feet is slightly higher than the value of 1.36 as determined for the *Jamestown* figures given

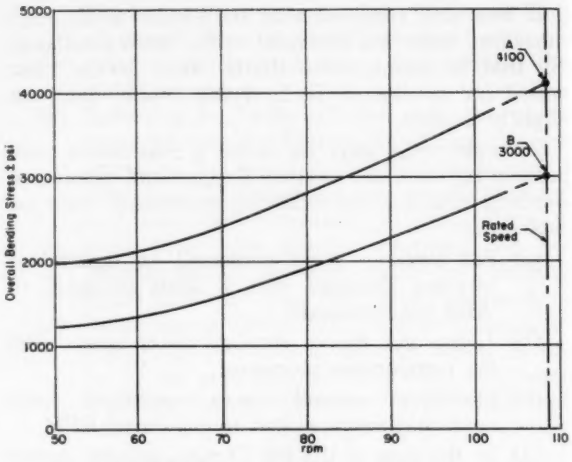


Figure 17. S.S. *Esso Jamestown.*
26,500 shp 108.5 rpm 19.75 knots 4 blades
Propeller Shaft Diameter 27⅛″ by A. B. S. 27″
Correction Factor: Size=Meg;ogob;e
Overall Bending Stress ± psi Calm Sea Condition
A—Light Displacement (Fig. 9)
B—Heavy Displacement (Fig. 3)

in Tables XI and XII. The individual ratios for light to heavy displacement are:

U.S.N.S. *Mission San Luis Obispo*	1.70
S.S. *Chryssi*	1.85
U.S.N.S. *Lt. James E. Robinson*	1.52
S.S. *Esso Jamestown*	1.36

It is apparent from these figures that there is no significant relationship between size or power of the vessel and the ratios of eccentricity factors for light and heavy displacement. It is suggested that the mean value of 1.6 might be considered representative and an upper value of 1.75 be considered suitable for estimation purposes for new designs.

SECTION VII. COMPARISON WITH RECOMMENDED FORMULAE

(a) Reference [6] was presented at the annual meeting of the S.N.A.M.E. held at New York on November 10 and 11, 1955. In a discussion to that paper it was proposed, that in addition to meeting the A.B.S. requirements, the section modulus of the tailshaft be no less than that calculated by the following expression:

$$S=C\frac{(M_g+M_t)}{6{,}000}$$

where:

S=section modulus $=\frac{I}{c}$

C=a factor of 1.75 to 2.0. This factor is used to compensate for extreme sea conditions, the omission of the moment due to torque reaction, the presence of stress concentrations

TABLE XIII

A.B.S. Shaft Diameters and Thrusts

Vessel	SHP	RPM	Knots	Shaft Diam.	A.B.S. Shaft Diam.	Thrust
Obispo	10,000	95	15.7	20.125	20.5	184,000
Chryssi	15,000	112	17.0	26.0	22.25	246,000
Robinson	8,500	85	17.55	21.0	20.5	156,000
Jamestown	26,500	108.5	19.75	27.125	27.0	385,000

TABLE XIV

Thrust Eccentricity—Light Displacement

Vessel	Thrust	Stress	$\frac{I}{c}$	e	Diam.	$\frac{e^*}{P}$
Obispo	184,000	4100	840	1.56	19.5	.080
Chryssi	246,000	7000	1035	2.45	22.0	.111
Robinson	156,000	5800	840	2.60	20.5	.125
Jamestown	385,000	4100	1935	1.72	23.0	.075

* Thrust Eccentricity factor, taken as a percentage of the propeller diameter.

TABLE XV

Thrust Eccentricity—Heavy Displacement

Vessel	Thrust	Stress	$\frac{I}{c}$	e	P	$\frac{e}{P}$
Obispo	184,000	2400	840	.91	19.5	.047
Chryssi	246,000	3750	1035	1.31	22.0	.060
Robinson	156,000	3750	840	1.67	20.5	.082
Jamestown	385,000	3000	1935	1.26	23.0	.055

and the addition of small alternating torsional stresses. Special consideration would be required for motor driven vessels when heavy alternating stresses, resulting from torsional vibration, are present.

M_g = gravity moment due to overhanging propeller weight calculated from forward face of the propeller.

M_t = calculated moment of eccentric thrust = .065 × Propeller Diameter × Rated Thrust.

6,000 = maximum safe fatigue limit to be used for the assembly operating in the presence of a corrosive medium. (psi)

Very rough calculations were made for the four vessels tested to date. A comparison of the resulting tailshaft sizes is tabulated as follows:

Vessel	Installed	A.B.S.	C=1.75	C=2.0
San Luis Obispo	20.125	20.5	22.375	23.25
S.S. *Chryssi*	26.0	22.25	26.0	27.25
Lt. J. E. Robinson	21.0	20.5	22.0	23.125
Esso Jamestown	27.125	27.0	30.0	31.5

A rough check of this formula, as applied to the S.S. *Esso Jamestown* indicates all measured stresses would be reduced in proportion to the increase in section modulus. Thus, when using the 1.75 factor, the measured stresses would be reduced by the ratio of 1955/2670, and when using the 2.0 factor, the measured stresses would be reduced by the ratio of 1955/3044. These ratios, 73.5 percent and 64 percent, would reduce the full power, light load, rough sea, anticipated stress of 8,000 psi, to 5,900 psi and 5,100 psi respectively. Heavy maneuvering conditions could still be expected to exceed 6,000 psi, but would not be expected to occur frequently enough as to cause particular concern.

(b) Reference [8] gives the most recent design procedures used by the U.S. Navy Dept., Bureau of Ships. Reference [9] describes the history of the Bureau of Ships design procedures.

It is apparent, that recent developments have prompted the Bureau of Ships to consider the problem of eccentric or off-center thrusts. Reference [8] says that the stress due to M_p shall not exceed 6,000 psi.

where: $M_p = M_g + M_{o.c.}$

and: M_g = gravity moment

$M_{o.c.} = 2M_g$ for single screw ships.

M_g in this case is equal to $W_p L_p$ where W_p is still the weight of the propeller assembly in water but

L_p is measured from the center of gravity of the propeller assembly to a point one shaft diameter forward of the after end of the strut bearing, for a wood or single rubber strip bearing.

If we assume that L_p is twice the distance from the strain gage position to the center of gravity, then we calculate the following shaft diameters:

Vessel	Shaft Diameter, inches
San Luis Obispo	21.125
S.S. *Chryssi*	26.0
Lt. J. E. Robinson	22.375
SS. *Esso Jamestown*	28.375

Admittedly these values are only approximate since all the necessary data were not available to make an accurate study. Nevertheless it does indicate close agreement between the recommended formula given in (a) above, with a c of 1.75, and the Bureau of Ships expression.

If we again apply the shaft diameter of 28.375 inches to the measured stresses obtained on the *Esso Jamestown*, the 8,000 psi stress referred to earlier is only reduced to 7,000 psi. Off hand, it would appear that the first expression would prove more reliable when applied to a large number of cases since it takes into account two important considerations:

(1) Although the nominal bending stress in a shaft will be higher at a point forward of the large end of the taper, the presence of stress concentrations and possible salt water at the end of the liner makes this area more susceptible to failure;

(2) The eccentric loading is primarily a function of thrust and not necessarily related to the gravity moment.

SECTION VIII. CONCLUSIONS AND RECOMMENDATIONS

The results of the work accomplished up to this time on single screw vessels, have clearly indicated:

1. That the alternating bending stresses in propeller shafts are by far the most serious dynamic loads on the shaft. This of course assumes no serious torsional criticals have been built into the system.

2. That the alternating bending stress which occurs at any particular time is principally the resultant of those components due to (a) eccentric thrust, (b) gravity moment, and (c) torque reaction.

3. That the magnitude of the resulting stress in the shaft is dependent on the actual phasing of these components. This in turn is primarily dependent on the draft of the vessel and the sea conditions.

4. Under the most adverse conditions of light displacement and rough seas the stress levels were observed to exceed by a factor greater than 2, the calculated stress obtained by combining in phase the three components mentioned above.

5. That the actual stress levels observed in service could explain the fatigue failures which have occurred.

Reviewing the results of fatigue studies carried out under this program and drawing heavily on the work of others, we may also note the following pointers:

1. Fatigue cracks can be expected to occur in fitted sections at stress levels of 5,000 psi or less. This was observed on tests conducted in air.

2. Although cracks can also be expected at the same stress levels in rolled shafts, the propagation of such cracks is greatly retarded by the rolling operation. Again the data were obtained on tests conducted in air.

3. Limited fatigue tests conducted in a corrosive medium indicate a marked decrease in fatigue strength to a level between 5,000 and 6,000 psi, regardless of the shaft composition.

4. Although the propagation of fatigue cracks is greatly retarded by the rolling operation, information is lacking on the actual gain to be expected when the assembly is subjected to the action of salt water over an extended period of time.

At this point it is rather obvious that the use of the existing American Bureau of Shipping formulae for sizing tailshafts is inadequate. This statement is based on the observation that all tailshafts measured to date may experience bending stresses in excess of 6,000 psi, under adverse operating conditions of light displacement and heavy seas. Since our experience indicates moisture in the propeller hub and fretting are strong possibilities, it is considered important that the bending stresses experienced in the tailshafts, be kept below 5,000 to 6,000 psi., the stress at which fatigue cracks originate.

The greatest difficulty in the development of suitable design procedures is associated with the estimation of the service stresses. In this regard there are two basic approaches possible. The more scientific method, and the one which would appeal to the theoretical man, would be the analytical approach. This would involve the complete development of all forces and moments from wake surveys and the proper combination of all stress components to determine the resultant shaft stress. An attempt to use this approach was carried out with fair results in the case of the U.S.N.S. Lt. *James E. Robinson*. This study and comparison with measured stresses was given in [5].

The second method is the empirical approach. This would of course be the most expeditious, more easily handled and consequently have greater appeal to the design and production engineers.

If we look at the overall problem we can readily see that the theoretical approach has serious limitations since it would only be possible to determine bending stresses accurately under fair weather conditions and it would still be necessary to estimate

the effects of weather which are considerable. Furthermore, the complexity of such a design procedure would be inconsistent with the American Bureau of Shipping approval procedures and the capabilities of some design offices. It would appear preferable therefore to develop an empirical method, possibly along the lines suggested in Section VII of this report.

We would recommend, as a design procedure, the use of simple empirical equations for determining tailshaft sizes. Such equations would however, be supported by adequate service testing and substantiated by theory.

As an example, it was shown in Tables XIV and XV, that the "Thrust Eccentricity Factor," e/P, varied considerably between the maximum and minimum value calculated. It is quite possible that this could be adequately explained by an analysis of the wake surveys and further correction factors developed for various hull configurations.

Our basic recommendations at this time may be summarized as follows:

(1) Tailshaft bending stresses should be limited to $\pm$6,000 psi under adverse loading and weather conditions;

(2) The potential advantages of cold rolling shafts in retarding the propagation of fatigue cracks, should be taken as an insurance factor;

(3) Empirical design procedures should be developed, based on service tests;

(4) Theoretical design procedures should be developed, to provide a better understanding of the problem and to provide correction factors for refinement of empirical expressions;

(5) To improve the reliability of emperical design procedure, further full scale testing should be carried out whenever possible, and is an absolute necessity for multiple screw vessels;

(6) Wake surveys, and calculated bending stresses should be obtained for all vessels tested and results correlated with test results;

(7) As an interim measure, new designs should be checked against the formula given in part (a) of Section VII.

REFERENCES

[1] "Strain Gage Installation for the Investigation of Tailshaft Bending Stresses on the S.S. *Esso Jamestown*," a report by Noonan, Knopfle & Feldman, 31 Dec. 1957.

[2] "An Experimental and Theoretical Investigation of Propeller Shaft Failures," by N. H. Jasper and Cdr. L. A. Rupp, U.S.N., Trans SNAME, 1952.

[3] "Propeller Shaft Stresses under Service Conditions—The S.S. *Chryssi* Investigation," by E. P. Panagopulos and A. M. Nicherson, Jr., Trans. SNAME, 1954.

[4] "Strain Gage Installation on the U.S.N.S. *Lt. James E. Robinson*," a report by Noonan, Knopfle and Feldman, 1 Dec. 1954.

[5] "Propeller Shaft Bending Stresses on the U.S.N.S. *Lt. James E. Robinson*," a report by Noonan, Knopfle and Feldman, 1 Sept. 1955.

[6] "Further Results from the Society's Investigation of Tailshaft Failures," by H. R. Neifert and J. H. Robinson, Trans. SNAME, 1955.

[7] "Propeller Shaft Stresses & Hull Vibrations of U.S.S. *Norfolk*, DL-1," Francis F. Vane & Anthony R. Paladino, DTMB report C-938. (To be published.)

[8] Navy Dept., Bu. Ships, Design Data Sheet DDS4301 by R. Michel.

[9] "Design of Naval Propeller Shafting as influenced by Full Scale Tests & Operating Experience," by R. Michel, to be published in ASTH Symposium on Full Scale Tests of Propeller Shafts. (1958)

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GIBBS & COX, INC.

NAVAL ARCHITECTS AND MARINE ENGINEERS

NEW YORK

DISCUSSION

of

NOTE ON BOTTOM OF PAGE 379

JOURNAL FOR MAY 1961

by

Mr. John H. Strandell

Propellor Group Bureau of Ships

Official Photograph, U.S. Navy

Propeller on Test Stand

Official Photograph, U.S. Navy

Damaged Port Propeller

Official U.S. Navy Photograph

Damaged Starboard Propeller

In reading the ASNE Journal issue May 1961 page 379 a quotation from the Marine Journal February 1961, the writer finds the information is not applicable to the present state of development. The Marine Journal claims that the controllable pitch propeller has limitations in its application, because of design difficulties with hub and blade designs in relation to blade width and number of blades, erosion, material, etc. These troubles have been overcome long ago and the writer feels that it's his duty to correct the misapprehension.

From the many lists of controllable pitch propeller installations in possession of the writer, there is all the indication that the application of c.p. propellers has no limitation as to size of c.p. propeller, and type and size of ships. There are large cargo ships, tankers, passenger ships, railroad ferries, oceanographic vessels, ocean tugs, trawlers, speed boats, patrol boats, fishing boats and all kinds of work boats in operation throughout the world equipped with c.p. propellers. These c.p. propellers are connected to all kinds of propulsion machinery: Reciprocating steam engines, steam turbines, gas turbines, diesel and gasoline engines and combinations thereof.

There are a great number of three (3) and four (4) blade Controllable Pitch Propeller designs in use. Several five (5) blade c.p. propellers have been designed and planned to be built. Hub sizes can always be kept to permissable diameters. Blade width has so far not limited the use of c.p. propellers. A Canadian high speed patrol boat uses two c.p. propellers absorbing a total of 5000 S.H.P. and they are provided with propeller blades that just leave enough clearance to pass each other when reversing. The same thing applies to a German Escort Destroyer that absorbs 38,000 SHP on two c.p. propellers. However, the most highly stressed and difficult c.p. propeller designed to date has been operating in a Swedish high speed boat since 1955. Two years ago the wide subcavitating propeller blades were replaced with supercavitating blades. This propeller is a three (3) blade design about 30 inches in diameter absorbing 1500 SHP at 2,000 RPM.

Much larger supercavitating c.p. propellers are being designed for 20,000 to 35,000 SHP using light weight blades made of titanium in order to reduce the large centrifugal forces.

The large physical size c.p. propeller is not necessarily a difficult design. For instance, the maximum working stress in some of the higher stressed parts of a 22 feet diameter 100 RPM, 20,000 SHP c.p. propeller have only half the working stress of the corresponding parts in a 12.5 feet diameter, 270 RPM, 20,000 SHP c.p. propeller.

A good example to verify that c.p. propellers can be made strong enough is shown in the three (3) pictures of a U. S. Landing Ship C. P. Propellers. One picture shows the c.p. propeller new on the test stand. The other two pictures show the Starboard and Port propellers after damage. The damaged blades were removed and replaced with new blades. No damage was inflicted to the hub structure or the inside mechanism. These c.p. propellers are 8'-8" in diameter absorbing 3,000 SHP each at 300 RPM.

The latest information from the many manufacturers in Europe is that they have more orders for c.p. propellers than they can take care of. (There are about seventeen (17) small and four (4) large manufacturers in Europe.) This rush of orders in itself should be an indication to the doubting Thomases that c.p. propeller application to ships has become a necessity in the minds of many Naval Architects, Marine Engineers and Ship Operators.

ASNE LOCAL CHAPTERS

FLAGSHIP SECTION
Baltimore, Md.-Washington, D. C. area
Charter granted—28 April 1961
Organized—20 June 1961

Officers:
Chairman: Mr. Uriah L. Allen, Jr., Westinghouse Electric Co.
Vice Chairman: Mr. Frank J. Smollen, Baltimore Shipbuilding and Dry Dock Co.
Secretary: Mr. Herbert F. Hathaway, General Electric Co.
Treasurer: Mr. Rupert A. Blue, Cutler-Hammer Co.
Councilors: 3 year term, Rear Admiral Charles M. E. Hoffman, USN (Ret);
2 year term, Mr. Irving G. Herman, Westinghouse Electric Co.;
1 year term, Captain Vernon E. Day, USCG.

The first ASNE local chapter to be formed was named FLAGSHIP SECTION at the request of its members. It was organized with 36 charter members by Mr. Uriah L. Allen, Jr. and its charter was presented to Mr. Allen by the President at the annual banquet as was announced in the may issue of the Journal.

The first meeting of the newly formed Flagship Section of the Society was held Tuesday evening, 20 June 1961, at the Westinghouse Electric Manufacturing Company Plant at Friendship Airport with a total of 32 persons present. The following 22 of the 36 charter members were in attendance:

Allen, Bauman, Beggs, Blue, Ciaramaglia, Day, Dodd, Donelson, Doremus, Elliott, Gerich, Hathaway, Herman, Kirby, Malinow, Radzykewycz, Reid. Sanford, Scharfenstein, Smollen, Suhrstedt and Wingquist.

The ten guests were:

Miss Evelyn Cerny, ASNE Headquarters; CDR Ward J. Davies, USCG CAPT J. E. Hamilton, USN (Ret), Secretary-Treasurer, ASNE; Mr. H. H. Hill, ASNE Council; LT David Kaetzel, USCG; RADM M. J. Lawrence, USN, President pro tem ASNE; Miss Ruth M. Leonard, ASNE Headquarters; CDR Ralph S. Lorimer, USNR, ASNE Council; LCDR H. B. Smith, USCG and CDR J. W. Thornbury, USN, Assistant Secretary-Treasurer, ASNE.

Following the very fine dinner, which had been preceded by a cocktail hour, in the Executive Dining Room, the Section was welcomed on behalf of Westinghouse, by Chairman-Elect, Uriah Allen. Captain Hamilton officially opened the meeting and announced the results of the election. Rear Admiral Lawrence installed Mr. Allen as Section Chairman, who in turn introduced the other three officers and the three members of the Section Council. Captain Day presented Mr. Allen with a beautiful gavel he had made of maple from Maine and walnut from Missouri with a stainless steel band engraved with organizational dates of the Society and the Section.

Mr. Allen called on the Section officers and members of the Council for brief comments.

As the first order of business, it was regularly moved and seconded that we accept the invitation of Captain Day and hold our next meeting at the U.S. Coast Guard Yard at Curtis Bay. Motion carried.

The Chairman urged active participation in the affairs of the Section by the members.

Each member was encouraged to offer suggestions for topics to be considered at future meetings.

The Treasurer, Mr. Rupert Blue, urged that each Charter member consider himself a member of the membership committee.

General comments were made and it was suggested that some of our meetings might be held in Washington.

Considerable optimism was registered concerning the possibilities of increasing our membership.

The Chairman stated he would call a meeting of Section officers and Council members within a few days so that a date for the next meeting might be established and future plans discussed.

The chapter Council held organizational meetings on 27 June 1961 and 18 July 1961. The chapters first technical meeting will be held in September at the Coast Guard Yard, Curtis Bay, Maryland.

OTHER CHAPTERS WHICH ARE AWAITING LEADERSHIP FOR FORMATION

Society members are available at the following locations, among others, in sufficient numbers to form local chapters. All that is needed is an organizer, someone who can visualize the potentialities of these local chapters and who will take the lead in signing up 19 other ASNE members to join with him in this endeavor.

Boston, Mass. Area
New York, N.Y. Area
Norfolk, Va. Area
San Diego, California Area
San Francisco, California Area

WHICH ONE WILL RECEIVE CHARTER NUMBER TWO?

FUEL CELLS

Edited by George J. Young
Published in 1960 by Reinhold Publishing Corporation
430 Park Avenue, New York 22, N. Y.
152 pages of text and illustration
Price $5.75

Reviewed by C. F. Viglotti
Head, Electrochemical Power Sources Section
Electrical Branch, Bureau of Ships

This book is based on the papers presented at a symposium held by the Gas and Fuel Chemistry Division of the American Chemical Society at the 136th National meeting in Atlantic City in September 1959.

The fuel cell is an electrochemical device in which chemical energy can be continuously converted to direct current electrical energy. The word "continuously" is the key word in this definition which sets fuel cells apart from primary and secondary batteries. In electrochemical batteries the energy which can be obtained before the battery is either discarded or recharged is limited by the quantity of active materials incorporated in the battery structure. The fuel cell is a convertor which will continuously deliver electrical energy so long as a fuel and oxidant are fed into the cell. A more descriptive name for the fuel cell is that it is a continuous feed primary battery.

In 1893, W. Grove first demonstrated the principle of the fuel cell. It was not until after World War II that any intensive research was made on fuel cells. In the interim the heat engine received much of the scientific effort in the energy conversion field. The efficiency limitations of heat engines and need for conserving fossil fuels has caused an upsurge in fuel cell research and developments.

In this volume workers in the field have contributed a group of papers covering the various fuel cell systems being currently investigated. The chapter headings which are listed below will illustrate the scope of these papers:

1. "Introduction," H. S. Liebhafsky and D. L. Douglas
2. "The Hydrogen-Oxygen (Air) Fuel Cell with Carbon Electrodes," Earl Kordesch.
3. "Catalysis of Fuel Cell Electrode Reactions," G. J. Young and R. B. Rozelle.
4. "Electrode Kinetics of Low-Temperature Hydrogen-Oxygen Fuel Cells," L. G. Austin.
5. "The High Pressure Hydrogen-Oxygen Fuel Cell," F. T. Bacon.
6. "High Temperature Fuel Cells," G. H. J. Broers and J. A. A. Ketelaar.
7. "Carbonaceous Fuel Cells," H. H. Chambers and A. D. S. Tentram.
8. "Nature of the Electrode Processes in Fuel Gas Cells," E. Gorin and H. L. Recht.
9. "Molten Alkali Carbonate Cells with Gas-Diffusion Electrodes," Daniel L. Douglas.
10. Summary of Panel Discussion.

Each paper presents a complete discussion of the respective fuel cell system including experimental results, performance and life data, details of construction, applications and future prospects. These papers may be considered as authoritative and possibly the best source for data and information on fuel cells since the authors are some of the major workers in the field.

Because these papers were prepared for presentation before a scientific group they will be of greatest value to those working in fuel cells and to the reader with a background in electrochemistry. This book is not recommended for the lay-reader who may have had his curiosity aroused by the articles on fuel cells which have recently appeared in some of the popular periodicals. It is, however, a book that fulfills a need in a field which is attracting the interest of many industrial research organizations.

BOOKS RECEIVED

"Operational Electricity" by Charles I. Hubert. A text for non-electrical engineers covering the theory, characteristics, applications and mode of operation of circuits and machines, published by John Wiley and Sons, Inc., 440 Park Avenue South, New York 16, N. Y. 530 pages, $8.50.

"Ship and Boat Builders Annual Review 1961," compiled by the editorial staff of *Ship and Boat Builder and Marine Trader*. A compilation of pertinent articles plus directories of ship and boat builders and marine equipment manufacturers, published by John Trundell Ltd., St. Richard's House, Eversholt Street, N.W. 1, London, England. 454 pages, 50 shillings.

"Marinas," second edition by Charles A. Chaney. Complete information and recommendations for design, construction and maintenance of modern water front facilities for recreational boats, published by National Association of Engine and Boat Manufacturers, Inc., 420 Lexington Avenue, New York 17, N. Y. 247 pages.

"Three Generations of Shipbuilding." A historical picture narrative of Newport News Shipbuilding and Dry Dock Company from 1886 to 1961, published by Newport News Shipbuilding and Dry Dock Company, Newport News, Virginia. 108 pages.

CHANGES IN MEMBERSHIP

The Society announces with pleasure that the following have joined its ranks since the publication of the May 1961 issue of the JOURNAL.

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Daria, Frank J.—From Naval to Junior

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Lord, Kenneth S. .....Associate

Von Dreel, William H., CAPT, USN (Ret) .....Naval

DEATHS

It is with deep regret that we announce that since the last JOURNAL, we have received notice of the death of:

Kennedy, Anthony, Jr., CAPT USN (Ret)

ROYAL NAVY LEADS IN COSAG MACHINERY

It is with considerable pleasure that we record a meritorious "first" by the Royal Navy, in establishing beyond doubt that the combined steam and gas turbine system is satisfactory for propelling major surface war vessels. The occasion was on April 23, when H.M.S. ***Ashanti***, Yarrow-built lead-ship of the seven ***Tribal***-class general-purpose single-screw frigates, completed contractors' trials.

We congratulate all concerned upon the successful fruition of upwards of seven year's sustained effort. As far as the machinery is concerned, the credit for this lies jointly with the Admiralty, the privately-owned Yarrow-Admiralty Research Department of Yarrow & Co. Ltd., A.E.I. Ltd. and the Fluidrive Engineering Co. Ltd. Y-ARD., it will be recalled, have been responsible for the co-ordination and design of the machinery spaces of all major post-war surface vessels of the Royal Navy; Associated Electrical Industries Ltd., whose Trafford Park branch, best known to our readers as Metropolitan-Vickers, have built the single-casing geared steam turbine and the 7,500 hp G6 gas turbine, and also the gearing, while Fluidrive have contributed the vital link which enables the non-reversing gas turbine to supplement the output of the steam turbine for short-term maximum speed, or to be run and manoeuvred independently for rapid getting under way in the face of some emergency.

Study has shown that a naval vessel of destroyer type develops over 20 per cent of its total installed power for rather less than four per cent of its service life. The steam plant therefore provides cruising power with considerable endurance and the gas turbine provides the necessary supplement for maximum speed. The fuel consumption of the gas turbine is, however, relatively high and it is thus only useful in conjunction with the more efficient prime mover. Its virtues of compactness, high power/weight ratio, quick starting and the ability to develop full output soon after a cold start are exactly the needed complement to the "base load" set and therefore result in a considerable saving of machinery space and weight. In this connection it is worthwhile drawing a parallel with the policy of the Central Electricity Generating Board, who have ordered for their Hams Hall power station a 15 MW Bristol-Siddeley Olympus gas turbo-alternator set for peak load "topping".

H.M.S. ***Ashanti***, is no one-off prototype, for the Royal Navy has adopted this form of machinery for nearly all its important new construction. Not only are the remaining six of the present batch of these vessels to be so powered but the four guided-missile "destroyers" of the ***Devonshire***-class, of which two are at present fitting-out afloat, will have twin-screw, more heavily gas turbine-boosted, plant of this type. There, a 15,000 shp A.E.I. steam turbine is backed up by similar A.E.I. G6 gas turbines, of 7,500 shp, so doubling the effective power which can be applied to the propeller shaft.

As we have said, ***Ashanti*** is the first large vessel in which this power concept has been applied, but we must record that a class of six twin-screw escort frigates is under construction in Hamburg for the Federal German Navy, and in these ships the normal cruising power, up to a maximum of 12,000 bhp, is developed by twin sixteen-cylinder M.A.N. 24/30 diesel engines on each shaft. These are backed up by a 13,000 hp Brown Boveri (Mannheim) gas turbine on each shaft. Whereas ***Ashanti*** has a 60 per cent power increase supplied by gas turbine, in the German vessels the gas turbine develops over twice the aggregate of the "basic" machinery. It may be noted, in passing, that the 9½ in bore by 11¾ in stroke diesel engines of the German frigates are among the most highly-rated applied to service afloat, with a bmep of over 220 lb/in² and a firing pressure of about 1,750 lb/in².

The booster gas turbine concept is not by any means new, for as long ago as February 1939 Dr. Meyer of Brown Boveri delivered a paper to the Institute of Mechanical Engineers in which he proposed alternative diesel and gas turbine machinery for a destroyer. This proposal was for 18,000 shp per shaft with a total machinery space length of 120 ft. The specific machinery weight of 20 lb/shp and the fuel consumption was 0.84 lb/shp-h at full power and 0.4 lb/shp-h when cruising on diesel engines. It was, however, only a proposal and we are glad that practical demonstration has taken place in this country.

H.M.S. ***Ashanti*** is the first of the second generation of post-war surface vessels of the Royal Navy and is officially known as a Type 81 first-rate (destroyer-type) frigate. All seven of her class will bear ***Tribal***-class names which have been honoured in two wars. She is 360 ft in length, 42 ft 6 in in beam and has a full load displacement of 2,700 tons. The armament is to consist of two 4.5 dual-purpose guns in single turrets, one forward and one aft, two twin-mounting 40 mm Bofors anti-aircraft guns, six 21 in torpedo tubes and a Limbo three-barrelled anti-submarine weapon. In addition there are to be two Seacat ship-to-air guided missile launchers and a helicopter will also be carried for A/S reconnaissance. H.M.S. ***Ashanti's*** forthcoming commissioning will bring new interest to the Royal Navy and fresh laurels to the engineering branch.

—THE MARINE ENGINEER AND NAVAL ARCHITECTS
May, 1961

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Westinghouse	3 Gateway Center, Pittsburgh 30, Penna.	508-9
C. H. Wheeler/Griscom-Russell	Philadelphia 32, Penna.	488
Whittelsey, Newton	17 Battery Place, New York 4, N.Y.	534

MARINE STEAM TURBINES

Steam turbine design appears to be operating under the law of diminishing returns. By taxing his ingenuity to the utmost the designer may obtain a 1% improvement in turbine efficiency and, even in cases where this can be achieved, he has to weigh carefully the factors of increased costs resulting from either changes in material or departures from present manufacturing techniques or a combination of both. In a paper on the subject of A. F. Veitch, read at a recent meeting of the North East Coast Institution of Engineers & Shipbuilders at Newcastle upon Tyne, Engl., the author pointed out that in contrast with land turbines now operating under steam conditions of some 3 million lb of steam per hour, the power requirements for merchant marine propulsion have increased but slightly —the average set now being of about 13,000 shp against 8000 shp about 8 years ago. This trend has been accomplished by a rise in average steam conditions of from 500 lb/sq. in. gauge 800°F to 600 lb/sq. in. gauge 900°F.

—MARINE JOURNAL
April, 1961

U.S. HYDROFOIL STUDY

In January, 1961 the Maritime Administration contracted with the Stanford Research Institute for a review of commercial hydrofoil economics and an evaluation of possible trade routes. This study is approximately at its mid-point. It is considering "first generation" ocean-going hydrofoil vessels, and is limited to vessels with maximum speeds of 60 knots. The largest would have gross weights of 500 tons. The upper limit of range without refueling would be between 500 and 1,000 nautical miles. The study should provide the prospective operator of hydrofoil vessels with the costs involved. The Maritime Administration's work for the last four years has been directed to providing answers to the technical and economic questions involved in designing and constructing commercial sea-going hydrofoil vessels, in conjunction with the basic work sponsored by the Office of Naval Research. It is hoped that within the next 6-8 months, enough data will have been accumulated to permit the design of a commercial vessel larger than the ***Denison*** for a particular service. The experience accumulated to date provides a sound basis for believing that hydrofoil vessels up to 500 tons GR and 60-knot speeds can be built and operated with confidence. Beyond this size and speed, more experience and study are needed. Economically, the outlook is not as clear for larger commercial hydrofoils.

—MARINE JOURNAL
April, 1961

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AUGUST, 1961

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